AS 1210—2010
(Incorporating Amendment Nos 1 and 2)

AS 1210—2010

Australian STANDARD

Pressure vessels

STANDARDS Australia

This Australian Standard® was prepared by Committee ME-001, Pressure Equipment. It was approved on behalf of the Council of Standards Australia on 28 July 2010.
This Standard was published on 19 October 2010.

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This Standard was issued in draft form for comment as DR 06703.

Standards Australia wishes to acknowledge the participation of the expert individuals that contributed to the development of this Standard through their representation on the Committee and through the public comment period.

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AS 1210—2010
(Incorporating Amendment Nos 1 and 2)

Australian Standard®

Pressure vessels

Originated in part as AS B31—1931 and AS CB1—1931.
Previous edition AS 1210—1997, AS 1210 Supplement 1—1990
and AS 1210 Supplement 2—1999.
Revised, amalgamated and designated AS 1210—2010.
Reissued incorporating Amendment No. 1 (November 2013).
Reissued incorporating Amendment No. 2 (July 2015).

Published by SAI Global Limited under licence from Standards Australia Limited, GPO Box 476, Sydney, NSW 2001, Australia

ISBN 978 0 7337 9687 6

PREFACE

This Standard was prepared by the Australian members of Joint Standards Australia/Standards New Zealand Committee ME-001, Pressure Equipment, to supersede AS 1210—1997, *Pressure vessels*, AS 1210 Supplement 1—1990 and AS 1210 Supplement 2—1999. This Standard is referenced in AS/NZS 1200, which is the parent Standard for pressure equipment and outlines general requirements for boilers, pressure vessels, pressure piping and related matters.

This Standard incorporates Amendment No. 1 (November 2013) and Amendment No. 2 (July 2015). The changes required by the Amendments are indicated in the text by a marginal bar and amendment number against the clause, note, table, figure or part thereof affected.

The issues discussed in Rulings RUL PE.3, RUL PE.4 and RUL PE.9 to PE.14 have been addressed in this revision, and those rulings will be withdrawn.

After consultation with stakeholders in both countries, Standards Australia and Standards New Zealand decided to develop this Standard as an Australian Standard rather than an Australia/New Zealand Standard.

The main changes in this edition are as follows:

(a) Incorporation of Amendments 1 to 3 to AS 1210—1997.

(b) Incorporation and review of Supplement 1 to AS 1210—1997 as Appendix H (on stress classification and limits), Appendix I (on finite element analysis) and Appendix M (on design against fatigue).

(c) Incorporation and review of Supplement 2 to AS 1210—1997 as Appendix L (on cold-stretched vessels).

(d) Revision of requirements for low temperature service, mainly in Clause 2.6.

(e) Revision of design tensile strength Table B1 (previously Table 3.3.1).

(f) Revision of Appendix A on design tensile strengths to align more closely with international practice.

(g) Revision of application of safety factors for flanges and transportable vessels.

(h) Deletion of the 400 mm manhole size from Table 3.20.9.

(i) Revision of Appendix E on information to be supplied to the designer.

(j) New Appendix G on failure modes.

(k) New Appendix J on wind and seismic loadings.

(l) New Appendix N on local non-pressure loads.

Minor changes have been made in the welding procedure, test plate, and postweld heat treatment requirements, principally to align with world practice. It is not intended that welding procedures already qualified will be invalidated by these changes or that the changes be applied retrospectively.

Amendment No. 2 to the 1997 edition reduced the factor of safety used to determine the material design stress from 4 to 3.5. This change is now confirmed in the Standard. The justifications for such a change included improvement in the quality of materials, improvement in the quality of welding and fabrication, improved inspection technology and better information on design, operation, maintenance and vessel failures.

Where other Standards refer to Supplements 1 and 2 to AS 1210—1997, this should now be taken as referring to this edition of AS 1210.

Statements expressed in mandatory terms in notes to tables and figures are deemed to be requirements of this Standard.

The terms ‘normative’ and ‘informative’ have been used in this Standard to define the application of the Appendices. A ‘normative’ appendix is an integral part of this Standard and an ‘informative’ appendix is only for information and guidance.

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FOREWORD

Design, manufacturing and supply requirements: The requirements in this Standard are intended to provide reasonably certain protection of life and property, and to indicate where a margin for deterioration in service may be needed to give reasonably long, safe equipment life. The Standard takes into consideration advancements in design and materials, and the evidence of experience.

The Standard contains basic data necessary for design, including material specification, design parameters, requirements for fabrication, testing and inspection. These requirements are specified in terms of principles where possible, with further detail added for uniform interpretation of principles and guidance on best methods. In other areas the Standard indicates where caution is necessary but a direct prohibition would be unwise at the present level of local and international knowledge.

In principle this Standard follows other codes forming part of AS/NZS 1200 in giving guidance to designers, manufacturers, inspection bodies, purchasers and users in the form of minimum engineering requirements that are necessary for the safe design, manufacture and testing of pressure vessels. In special instances additional requirements may be necessary for adequate performance or safety.

The Standard classifies vessels based on different classes of construction, and gives basic principles to indicate where such classes should or are to be used. Four classes (1, 2A, 2B, 3) use a stress safety factor of 3.5, and four classes (i.e. 1H, 2H, and 1S and 2S using cold-stretching) use higher design stresses.

No rules for design and manufacture can be written in sufficient detail to ensure good workmanship in manufacture. Each vessel manufacturer is responsible for taking every necessary step to make sure the quality of manufacture is such as will ensure compliance with good engineering practice and design.

The user of pressure vessels will also need to consider many factors beyond those covered by this Standard in the final specification of a vessel and is cautioned that the Standard is not a complete design handbook and there is a need for competent engineering judgement.

Adaption for regulatory change: The Standard continues to be written largely for Australian conditions and to cater for recent moves in various States and Territories to objective or performance regulations rather than the earlier prescriptive ones. These moves also have lead to privatization of inspection functions such as design verification, manufacture and in-service inspection, and agreement by designers, manufacturers, purchasers and others involved.

Thus the Standard uses competent ‘inspection bodies’ (somewhat like ‘notified bodies’ in European practice) in place of the previous regulatory authority and is written as far as practical for clear interpretation and use in contracts to assist all parties and facilitate safety and trade.

Use of alternative methods: In addition to the flexibility provided for the various classes of vessels, provision is made for the use of specific alternative methods, materials, and the like where equivalent safety and performance is achieved and any departures from the Standard are clearly identified in all documentation and are agreed.

Use of international Standards: Acknowledgment is gratefully made to the American Society of Mechanical Engineers for permission to reproduce certain extracts from the *ASME Boiler and Pressure Vessel Code*. In addition, acknowledgment is made of the considerable assistance provided by British and other national Standards, recent EN Standards and the recent draft ISO Standards for pressure vessels. This takes advantage of world experience as well as Australian experience, and helps to align with these major Standards to optimize safety and standardization and facilitate trade.

Compliance with the appropriate class of this Standard will satisfy the technical requirements for equivalent vessels to the above national, regional and international Standards. However, compliance with regulatory and quality requirements of the country of use will need to be satisfied. For comparison of this Standard with the above Standards see AS/NZS 1200.

Effect on existing designs of vessels: The revised and new requirements in this edition of the Standard are not intended to require modifications to any existing vessels that were constructed to previous editions of the Standard. However, there may be cases where implementing the new requirements to an existing design could be advantageous. Existing designs for future vessels should be suitably revised.

STANDARDS AUSTRALIA

Australian Standard
Pressure vessels

SECTION 1 SCOPE AND GENERAL

1.1 SCOPE

This Standard sets out minimum requirements for the materials, design, manufacture, testing, inspection, certification, documentation and dispatch of fired and unfired pressure vessels constructed in ferrous or non-ferrous metals by welding, brazing, casting, forging, or cladding and lining and includes the application of non-integral fittings required for safe and proper functioning of pressure vessels. This Standard also specifies requirements for non-metallic vessels and metallic vessels with non-metallic linings.

For detailed requirements for metallic materials, manufacture, testing and conformity assessment, reference is made to the relevant material Standards, AS 4458, AS 3920.1, AS/NZS 3992 and AS 4037.

The requirements of this Standard have been formulated on the basis that the required examinations and inspection during manufacture are performed and that appropriate care is taken during subsequent stages in the life of vessels. Appropriate care may include transport, installation (for guidance see AS 3892), operation and maintenance (for guidance see AS 3873), and in-service inspection (for guidance see AS/NZS 3788). Other standards or procedures may also be relevant.

Users of this Standard are reminded that it has no legal authority in its own right, but may acquire legal standing in one or more of the following circumstances:

(a) Adoption by a government or other authority having jurisdiction.

(b) Adoption by a purchaser as the required standard of construction when placing a contract.

(c) Adoption where a manufacturer states that a vessel is in accordance with this Standard.

1.2 OBJECTIVE AND PERFORMANCE CRITERIA

1.2.1 Objective of the Standard

This Standard aims to specify clear, uniform, safe requirements that—

(a) cover the materials, design, manufacture, testing, inspection, certification, documentation and dispatch of pressure vessels; and

(b) facilitate the supply of pressure vessels which meet the purchaser's requirements.

1.2.2 Performance requirements

To meet the above objective, pressure vessels supplied to this Standard should satisfy the following performance criteria when produced and used in accordance with the contract, the designed service conditions and sound practice:

(a) Provide reasonably certain protection of all persons involved in various stages of the vessel's life and of adjacent property and environment.

(b) Provide appropriate economy, performance, reliability, operability, inspectability and maintainability over a reasonably long life.

(c) Control risks to at least satisfy applicable safety, health and environment laws.

(d) Comply with the safety recommendations of AS/NZS 1200, Appendix J.

NOTE: See Appendix ZZ for a comparison of the requirements of this Standard with Appendix J of AS/NZS 1200 and with ISO 16528-1.

The remainder of this Standard gives prescriptive requirements that satisfy the above criteria in the matters covered.

NOTE: As the competence of construction bodies and personnel is integral to satisfying the objectives of this Standard, guidance is given in Appendix P.

1.3 APPLICATION

This Standard is intended to apply to pressure vessels—

(a) with design pressures above the curves in Figures 1.3.1 and 1.3.2 for welded, forged, brazed or cast metallic vessels or non-metallic vessels unless otherwise agreed by the parties concerned; and

(b) with operating temperatures within the temperature limits for various materials and components as stated in the appropriate Section of this Standard.

In relation to pressure-containing parts, the following shall be included in the scope of this Standard:

(i) Where external piping is to be connected to the vessel—

  (A) the welding end connection for the circumferential joint for welded connections;

  (B) the first threaded joint for screwed connections;

  (C) the face of the first flange for bolted, flanged connections; and

  (D) the first sealing surface for proprietary connections or fittings.

(ii) Where a non-pressure part is attached directly to either the internal or external surface of a pressure vessel—

  (A) the weld attaching the part to the vessel; and

  (B) lifting lugs, support rings, straps and attached load-carrying brackets or cleats for items such as platforms, pipe supports, manhole davits, process internal supports, etc.

(iii) Pressure-retaining covers for vessel openings such as manhole and handhole covers.

(iv) Vessel supports, legs and skirts that form part of the vessel.

(v) Protective devices, pressure relief valves and thermal protection where required by the purchaser.

The scope for attachments does not include items such as cranes, walkways, platforms etc., however any loads applied to the vessel by such items shall be considered in the vessel design.

This Standard is not intended to apply to liquid storage tanks, large low pressure gas storage tanks (such as are dealt with in API 620), nuclear vessels, machinery such as pump and compressor casings, or vessels subject to pressures caused only by static head of their contents, fire-tube, shell, water tube and miscellaneous boilers, water tube boilers, non-integral piping, and other plant under pressure excluded by AS/NZS 1200.

Related Standards that provide alternatives to the requirements in this Standard within the scope of their application are AS 2971 and AS/NZS 3509.

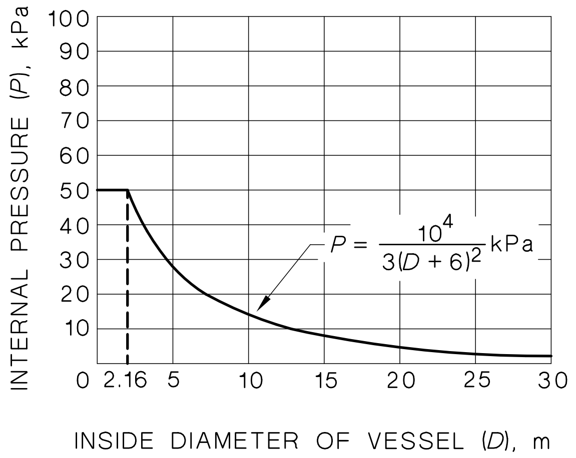

FIGURE 1.3.1 VESSELS SUBJECT TO INTERNAL PRESSURE

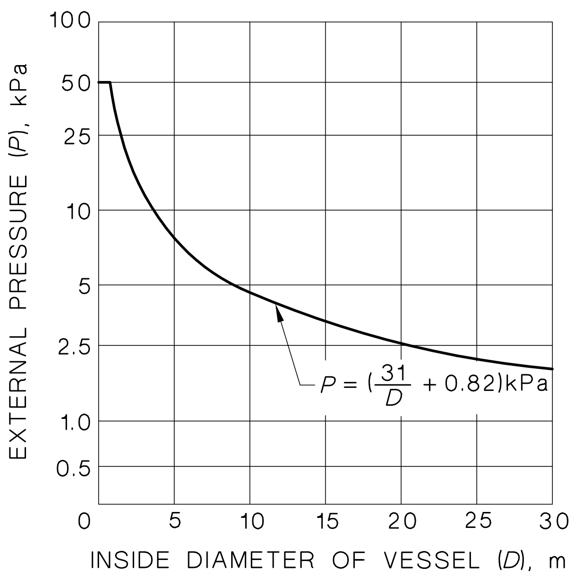

FIGURE 1.3.2 VESSELS SUBJECT TO EXTERNAL PRESSURE

1.4 INTERPRETATION OF STANDARD

For interpretation of this Standard refer to AS/NZS 1200.

1.5 NEW DESIGNS, MATERIALS AND MANUFACTURING METHODS

This Standard does not prohibit the use of materials or methods of design or manufacture that are not specifically referred to herein. (See AS/NZS 1200 for guidance).

1.6 CLASSES OF VESSEL CONSTRUCTION

Metallic vessels are classified according to the design, manufacture, testing and inspection requirements indicated in Table 1.6. Class 2 is subdivided into classifications 2A and 2B to enable the use of higher weld joint efficiency where spot non-destructive examination is used in addition to a production test plate.

For mixing of classes of welded construction, see Clause 1.7.2.

Cast iron and non-metallic vessels are not classified.

The range of materials permitted for vessels of Classes 1H, 2H, 1S and 2S construction (which permit higher design strength values) is limited by Clause 2.1.1.

The extent of non-destructive examination may be reduced from that required for Class 1H construction (see AS 4037) provided that criteria for design against fatigue failure, as appropriate for Class 2H, are fulfilled (see Appendix M).

TABLE 1.6

VESSEL CLASSIFICATION—SUMMARY (see Notes 1 and 2)

Item No.	Description and clause reference	Requirement of Class							
	Class	**1H**	**2H**	**1S**	**2S**	**1**	**2A**	**2B**	**3**
	General description	**High stress, full NDE**	**High stress, part NDE**	**Cold stretched, full NDE**	**Cold stretched, part NDE**	**Medium stress, full NDE**	**Reduced stress, part NDE**	**Reduced stress, no NDE**	**Low stress, no NDE, no WPTP**
1	**CLASSES** (Clause 1.7 and Table 1.7) Note 4								
2	**MATERIALS** (Section 2)								
2.1	**Steel: plate, strip, forging and casting (for weldability see Table 1.7B)**								
	–Pressure quality	Any	Excellent weldability	Austenitic stainless steel, e.g. Types 304, 316		Any	Good weldability		Excelent or good weldability
	–Structural quality	Not permitted		Not permitted		Any listed	Good weldability		Good weldability
2.2	**Non-ferrous metals – pressure quality**	Any		Not permitted		Any	Any		Any
2.3	**Cast iron**	*Not classified*							
2.4	**Non-metallic–pressure quality** (See Section 10)	*Not classified*							
3	**DESIGN** (Section 3)								
3.1	**Design methods** (Clause 3.1.3)								
	–by formula	Any (Clauses 3.7 to 3.32)							
	–by analysis	Any (Appendix H and I)							
	–by experiment	Any (Clause 5.12)							
	–by fracture mechanics	Any (Clause 3.1.3)							

(*continued*)

TABLE 1.6 (*continued*)

Item No.	Description and clause reference	Requirement of Class							
	Class	**1H**	**2H**	**1S**	**2S**	**1**	**2A**	**2B**	**3**
	General description	**High stress, full NDE**	**High stress, part NDE**	**Cold stretched, full NDE**	**Cold stretched, part NDE**	**Medium stress, full NDE**	**Reduced stress, part NDE**	**Reduced stress, no NDE**	**Low stress, no NDE, no WPTP**
3.2	**Design strength (*f*) at design temperature *T*** (Clause 3.3.1, Appendix A, B and L)								
3.2.1	Ductile metals other than structural quality and bolting. For bolting see Appendix A.	Below creep range, lowest of: $\frac{R_{m}}{2.35};\frac{R_{mT}}{2.35}\left(\text{or } \frac{R_{m}}{2.5};\frac{R_{mT}}{2.5} \text{ for austenitic steels}\right)$; $\frac{R_{e}}{1.5};\frac{R_{eT}}{1.5}$ In creep range, for indefinite life, lowest of: $\frac{S_{R}}{1.5};\ S'_{c}$ In creep range, for life of t hours, lowest of: $\frac{S_{Rt}}{1.3};\ S_{ct}$				Below creep range, lowest of: $\frac{R_{m}}{3.5};\frac{R_{mT}}{3.5};\frac{R_{e}}{1.5};\frac{R_{eT}}{1.5}$ In creep range, for indefinite life, lowest of: $\frac{S_{R}}{1.5};\ S'_{c}$ In creep range, for life of t hours, lowest of: $\frac{S_{Rt}}{1.3};\ S_{ct}$			
3.2.2	For Low ductility metals (A_5 <15%)	Lowest of Item 3.2.1 (below creep range only), and $R_{m}\left(0.1 + 0.023A_{5}\right)$				Lowest of Item 3.2.1 (below creep range only), and $R_{m}\left(0.1 + 0.023A_{5}\right)$			
3.2.3	Structural quality	Not permitted				Values as for Item 3.2.1, multiplied by 0.92			
3.2.4	Non-metallic (Section 10)	As agreed by the parties concerned with the proviso that the design complies with the relevant vessel design Standard (e.g. AS 2971).							
3.3	**Design quality factors**								
3.3.1	Joint efficiency (η) (longitudinal)								
	–welded (Clause 3.5.1.7 and Table 3.5.1.7)	1.0	1.0	1.0	1.0	1.0	0.85	0.80	0.7
	–brazed (Clause 3.5.3.4)	Not permitted		Not permitted for joints A, B or D in vessel shell		1.0	1.0 or 0.5 depending on examination results		
	–soldered (Clause 3.5.4)	Not permitted				1.0	1.0 or 0.5 depending on examination results		
3.3.2	Casting quality factor (Clause 3.3.1.1)								
	–C, C-Mn, low alloy and high alloy steels	0.8, or 0.9 with additional testing		Not permitted		0.9	0.8, or 0.9 with additional testing		
	—Non-ferrous and ductile cast iron	0.8, or 0.9 with additional testing		Not permitted		0.9	0.8, or 0.9 with additional testing		
	–Grey cast iron	Not permitted		Not permitted		1.0 (not permitted for lethal or flammable fluids)			
3.4	**Local load assessment**	Required (See Appendix N)				Not typically required (See Appendix N)			
3.5	**Fatigue assessment** (Clause 3.1.4 and Appendix M)	Required for >500 full pressure cycles or equivalent				Recommended for >50 000 full pressure cycles	Recommended for >100 000 full pressure cycles		
	NOTE: A full cycle is pressure from 0 to P to 0 or equivalent stress range								
3.6	**Accidental fire assessment**	Required (See Clause 8.6)				Not required			
3.7	**Collision assessment**	Required (See Appendix N)				Not required			

(*continued*)

TABLE 1.6 (*continued*)

Item No.	Description and clause reference	Requirement of Class							
	Class	**1H**	**2H**	**1S**	**2S**	**1**	**2A**	**2B**	**3**
	General description	**High stress, full NDE**	**High stress, part NDE**	**Cold stretched, full NDE**	**Cold stretched, part NDE**	**Medium stress, full NDE**	**Reduced stress, part NDE**	**Reduced stress, no NDE**	**Low stress, no NDE, no WPTP**
3.8	**Welded nozzle connections**								
	–Integral reinforcement (Clause 3.19.3)	Recommended		Recommended		As required by Clause 3.19			
	–Partial penetration welds	Analysis required		Analysis required		Permitted			
	–Threaded joints (Clause 3.19.4.2)	Permitted		Permitted		Permitted			
	–Brazed joints (Clause 3.5.3.1)	Permitted if design temperature ≤205°C		Not permitted (except for attached piping)		Permitted if design temperature ≤205°C			
	–Soldered joints (Clause 3.5.4)	Not permitted		Not permitted (except for attached piping)		Permitted if design temperature ≤50°C			
4	**MANUFACTURE AND WELDING (Note 3) (Sections 4 and 5, AS 4458 and AS/NZS 3992)**								
4.1	**Welding procedure qualification**	Required to AS/NZS 3992 (except as provided for in Item 4.2)							
4.2	**Prequalified welding procedure**	Permitted to AS/NZS 3992 but subject to partial requalification, e.g. welder qualification							
4.3	**Criteria for weld quality** (AS 4037)	Very high				High			Reduced
4.4	**Personnel requirements** (See Clause 4.2.2)								
4.5	**Postweld heat treatment** (AS 4458) (also may be required to satisfy service conditions)								
	–C, C-Mn steels	Required if t >32 mm, or as per AS 4458	Not required			Typically not required (See AS 4458)	Not required, except required for Class 2A transportable vessels		
	–Low alloy steels	AS 4458							
	–Quenched and tempered steels	AS 4458							
	–Austenitic steels	Not required							
	–Non-ferrous	Not required							
5	**EXAMINATION AND TESTING (See Section 5 and AS 4037)**								
5.1	**Visual**	100%		100%		100%			
5.2	**Penetrant (PT) or Magnetic particle (MT)**	0-100%		0-20%		0-10%	Not required		
5.3	**Radiographic (RT) or Ultrasonic (UT)** (AS 4037) — Weld: Long Circ.	100% 100%	100% 10% or 25% or 100%	100% 100%	100% 20%	100% 10% or 25% or 100%	2% or 10% 2% or 10%	Not required	
5.4	**Production test plates** (Clause 5.2 and AS/NZS 3992)	Required		Required		Required (some exceptions)	Required (some exceptions)	Required (limited testing)	Not required
5.5	**Hydrostatic test** (Clause 5.10)	Required (except as provided for in Items 5.6 or 5.7)							
5.6	**Pneumatic test** (Clause 5.11)	Permitted by agreement							

(*continued*)

TABLE 1.6 (*continued*)

Item No.	Description and clause reference	Requirement of Class							
	Class	**1H**	**2H**	**1S**	**2S**	**1**	**2A**	**2B**	**3**
	General description	**High stress, full NDE**	**High stress, part NDE**	**Cold stretched, full NDE**	**Cold stretched, part NDE**	**Medium stress, full NDE**	**Reduced stress, part NDE**	**Reduced stress, no NDE**	**Low stress, no NDE, no WPTP**
5.7	**Proof test** (Clause 5.12)	Required for special applications							
5.8	**Leak test** (Clause 5.13)	Required for special applications							
6	**CONFORMITY ASSESSMENT** (Section 6 and AS 3920.1)					Required			
7	**MARKING** (Section 7 and Clause 1.6)					Required			
8	**PROTECTIVE DEVICES** (Section 8)					Required			
9	**DISPATCH** (Section 9 and AS 4458) Required to be cleaned and protected for transport and storage as appropriate								

LEGEND:

See Appendix A for A_5, R_m, R_{mT}, R_e, R_{eT} notation.

t = nominal thickness above which postweld heat treatment is required, in millimetres
NDE = non-destructive examination
WPTP = welded production test plate

NOTES:

1 This Table summarizes the requirements of this Standard as they apply to each Class. Reference should be made to the text for full details.

2 Materials, design, welding, qualification and testing, marking, dispatch and non-metallic vessels are shown as ‘permitted’ on the basis that such items comply in all other aspects with this Standard.

3 Welding is taken to include brazing and soldering unless otherwise specified. For detailed requirements, see AS 4458 and AS/NZS 3992.

4 Riveted construction is not permitted for Classes 1H, 2H, 1S and 2S vessels.

1.7 APPLICATION OF VESSEL CLASSES

1.7.1 General

The application of various classes of construction shall comply with Clause 1.7.2 and Tables 1.7A and 1.7B, and with any necessary additional risk assessment requirements.

1.7.2 Mixed classes of construction

Mixing of classes of construction is permitted, provided the following conditions apply:

(a) The class of construction used for any part, component, or joint is not a lower class than that required by Tables 1.6, 1.7A and 1.7B, as applicable at the part or joint.

(b) Where two classes join at a circumferential joint, and the longitudinal joints of a vessel are fully radiographed, Type B (see Clause 3.5.1.1) circumferential joints shall be spot radiographed in accordance with the relevant requirements for the Clause 'spot examination' of AS 4037.

(c) The design, marking and manufacturer's data report record joint compliance.

(d) The vessel shall be marked with the highest two classes of construction used in the main vessel shell.

Examples of pressure vessels where mixed classes of construction may be used are—

(i) vessels having different sections exposed to different process conditions that warrant different classes of construction, e.g. major refinery towers and heat exchangers;

(ii) vessels having different wall thicknesses over the length of the vessel owing to external load considerations (e.g. wind or self-weight) or different diameters; and

(iii) Class 1 shell joined to a Class 1 end by a Class 2 weld that meets all the provisions and limitations for Class 2 construction.

TABLE 1.7A

APPLICATION OF VESSEL CLASSES FOR SERVICE CONDITIONS

		Service limit for vessel classes (see Note 1)									
Item No.	**Service condition**	**1H (Note 4)**	**2H**	**1S (Note 4)**	**2S**	**1 (Note 4)**	**2A**	**2B**	**3**	**Low ductility $A_5 < 5\%$ (Note 5)**	**Non-metallic**
1	**Fluid type** (see AS 4343)										
	—Lethal (Notes 2 and 3)	No limit	Not permitted	No limit	Not permitted	No limit	Not permitted			Not permitted	As agreed
	—Other fluid types (Note 3)	No limit		No limit		No limit				No limit	
2	Design pressure (Clause 1.3)	No limit								See Notes to Table B1(c)	
3	**Design temperature**										
	—Maximum (Clause 2.7)	No limit, except for material selection and stress		400°C		No limit, except for material selection and stress					
	—Minimum (Clause 2.6)	No limit, except for material selection, stress and postweld heat treatment									
4	**Shell thickness**	Thickness of type A or B butt welds in Figure 3.5.1.1									
	—Maximum (Table 1.7B)	No limit	Table 1.7B	30 mm		No limit		Table 1.7B			As agreed
	—Minimum (Clause 3.4)	Table 3.4.3									
5	**Shell diameter**	No limit									
6	**Corrosion/erosion**	No limit, except for material selection, fluid type, stress and corrosion allowance									
7	**Fatigue service** (Clause 3.1.4 and Appendix M)	Required for >500 full pressure cycles or equivalent				Recommended for >50 000–100 000 pressure cycles				Not normally required	As agreed
8	**Transportable vessels, service** (Clause 3.26.3)	No limit provided designed against fire, collision and fatigue. Not permitted for Group G steels	Not permitted	No limit provided designed against fire, collision and fatigue	Not permitted	No limit	See Clause 3.26.3.1		Not applicable	500 L max. with non-harmful fluid	

NOTES:

1 Applicable to welded, brazed, soldered, seamless, forged, cast metallic vessels and non-metallic vessels.

2 Vessels shall be forged, seamless or Class 1H or Class 1 construction.

3 Packed floating heat exchangers shall not be used when the fluid in contact with the joint is lethal or flammable, unless a risk assessment proves it to be satisfactory.

4 Classes 1H, 1S and 1 construction shall be used where it is not practicable to provide inspection openings for subsequent inspections (Clause 3.20.6(b)).

5 See Appendix A for A_5.

TABLE 1.7B

NOMINAL SHELL THICKNESS REQUIRING VARIOUS CLASSES OF CONSTRUCTION (Note 1)

Material (Notes 7 and 9)			Nominal shell thickness (Note 2)		
Group	Type	Typical standard or nominal composition	Class 1 & 1H construction mm	Class 2 & 2H construction mm	Other Classes
A1	Carbon and carbon-manganese steel (low strength)	AS 1548: 7-430, PT460	>32 (Note 3)	>20	
A2	Carbon and carbon-manganese steel (medium strength)	AS 1548: 5-490, 7-490	>32 (Note 3)	>12	
A3	Carbon and carbon-manganese steel (high yield strength)	AS 1548: PT490, PT540 AS/NZS 1594: XF 400, XF 500 API 5L: X52, 60, 65, 70	>32 (Note 3)	>20	
A4	Carbon and carbon-manganese steel (quenched and tempered)	JIS-G 3115 SPV490	>32	>12	
B	Alloy steel (alloy <¾)	C-½ Mo; ½ Cr-½ Mo; 1¼Mn-½Mo	>20	>10	
C	Alloy steel (¾ ≤ total alloy <3)	1 Cr-½ Mo; 1¼ Cr-½ Mo	>16	>6	
D1	Low alloy steel (vanadium type)	½ Cr-½ Mo-¼ V	All	—	
D2	Alloy steel (3 ≤ total alloy <10)	2¼ Cr-1 Mo; 5 Cr-½ Mo; 9 Cr-1Mo	All	—	
E	3½–5 Nickel steel	3½ Ni, 5Ni	>16	>6	
F	9 Nickel steel	9 Ni	All	—	
G	Alloy steel quenched and tempered	AS 3597: 700 PV	All	—	
H	Martensitic chromium steel	12 Cr (Type 410) 15 Cr (Type 429)	All	—	See Notes 10 and 11
J	Ferritic high chromium steel	12 Cr-Al (Type 405) (Note 4)	All	—	
		12 Cr-low C (Type 410S) (Note 5)	All	—	
		12 Cr-low C (Type 410S) (Note 6)	>38	>5	
K	Austenitic chromium-nickel steel	18 Cr-8Ni (Type 304) 18 Cr-12Ni-2.5 Mo (Type 316) 18 Cr-10Ni-Ti (Type 321)	>38	>10	
L	High chromium steel (>25 Cr)	27 Cr-0.5Ni-0.2C (S44627)	All	—	
M	Ferritic-austenitic chromium-nickel steel (Duplex and Super Duplex)	Various	>38	>5	
Non-ferrous metals	Aluminium and its alloys	Various	>12	≤12	
	Copper and its alloys	Various	>6	≤6	
	Nickel and its alloys	All grades except those below	>38	>5	
		Ni-Cr-Fe, Ni-Fe-Cr, Ni-Mo, Ni-Mo-Cr, Ni-Cr-Mo-Nb	>10	≤10	
	Other	Various	Note 8	Note 8	

* See Notes page 19.

NOTES TO TABLE 1.7B:

1 This Table does not prevent Class 1 and 1H or Class 2 and 2H construction below thickness shown. However, it nominates the minimum thickness over which these constructions must be used.

2 See also Clause 1.7, and for clad plate, see Clause 3.3.1.2.

3 This may be increased to 40 mm where a preheat of not less than 100°C is used or the steel used is made to fully killed fine grain practice with longitudinal impacts of 27 joules at −20°C.

4 Welded with straight chromium welding consumables.

5 Welded with any welding consumable other than in Note 6.

6 Welded with welding consumables that produce an austenitic chromium-nickel steel weld or a non-hardening nickel-chromium-iron deposit.

7 For basis of grouping of steels, see AS/NZS 3992 and for specific materials, see Table B1.

8 By agreement between the parties concerned.

9 Excellent weldability = Groups A1, A2, A3, A4 and K. Good weldability = Groups B, C, E and M.

10 For Classes 1S and 2S the maximum nominal shell thickness = 30 mm.

11 For Class 3 construction the nominal shell thickness shall be less than that for Class 2.

1.8 DEFINITIONS

For the purpose of this Standard, the definitions in AS 4942, AS 2812 (for welding terms and symbols) and the following apply.

1.8.1 Actual thickness

The actual measured thickness of the material used in the vessel part. It may also be taken as the nominal thickness, minus any applicable manufacturing under tolerance (see Clause 3.4.2(i)).

1.8.2 Cold-stretched plate

Plate, sheet or strip cold-stretched in a cold-stretching machine, after solution heat treatment, to a controlled 0.2% residual proof strength e.g. 350–450 MPa (or 400–500 MPa for nitrogen-alloyed steels).

1.8.3 Cold-stretched pressure vessel

A pressure vessel subjected to a calculated and controlled internal pressure (cold-stretching pressure) at ambient temperature during hydrostatic testing. This pressure stretches the steel, as necessary, and within limits to raise its proof strength and to ensure the actual thickness × proof strength satisfies the design. Also known as a ‘pressure strengthened vessel’.

NOTE: Vessels made of cold-stretched or work hardened plate will normally have very little plastic deformation during cold-stretching.

1.8.4 Construction

All of the main stages in the provision of pressure vessels. See Figure 1.8.4.

It includes all the tasks required to make and supply the vessel, excluding the specification, and including (but not limited to) design, selection and supply of materials or components, fabrication, provision of examination, testing and conformity assessment services.

NOTE: Any of these tasks might be provided by different parties, as agreed in the contract.

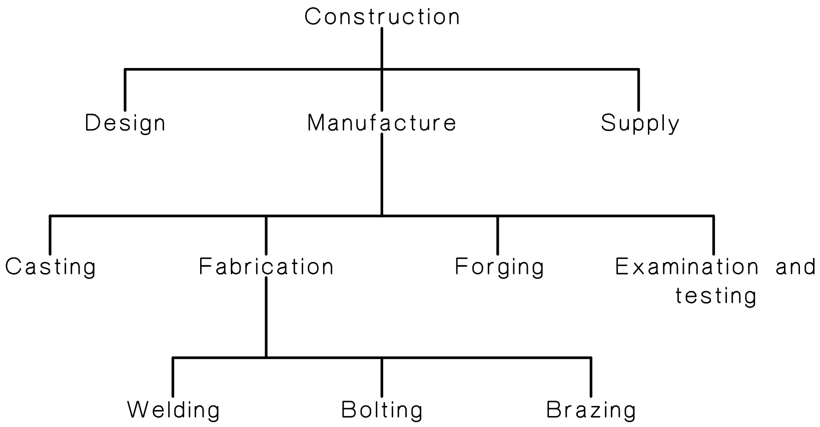

FIGURE 1.8.4 TERMS USED IN CONSTRUCTION

1.8.5 Design

Drawings, calculations, specifications, reports, purchaser requirements and all other information necessary for the complete description of the vessel and its manufacture.

1.8.6 Design life

The life specified in the design as agreed between the manufacturer and the purchaser for each component of pressure equipment.

NOTES:

1 Design life is expressed in—
   (a) hours (or nominated as 'indefinite'), for operating in the creep (high temperature) range;
   (b) number of cycles, for fatigue; or
   (c) years, for corrosion or other material degradation.

2 The design life relates to the performance of the relevant component and is not necessarily related to the life of the pressure equipment.

1.8.7 Design pressure

The maximum gauge pressure, at a designated temperature, which is allowed at the top of the vessel in its operating position. (Also known as 'maximum allowable working pressure'.)

1.8.8 Design strength

The maximum allowable stress for use in the equations for the calculation of the minimum thickness or dimensions of pressure parts (see Clause 3.3).

1.8.9 Design temperature

The metal temperature at the coincident calculation pressure used to select the design strength for the vessel part under consideration (see Clause 3.2.2).

1.8.10 Fire condition

An unusual but foreseeable service condition where there is a significant hazard to a vessel due to exposure to a fire, or other similar source of heat.

The hazard may exist due to flammable or combustible contents, or where such materials are present in the surrounding area.

1.8.11 Harmful contents

A substance, which under the expected concentration and operating conditions, is classified as a *combustible liquid* or *fluid irritant* to humans, or is harmful to the environment, above 90°C, or below –30°C. (See AS 4343.)

1.8.12 Hazard level

A rating of the hazard associated with an application, as determined using AS 4343.

1.8.13 Ligament efficiency

The ratio (expressed as a decimal) of the lowest calculated working strength of the ligaments between holes, for any way in which any ligament might fail, to the calculated working strength of the solid plate adjacent.

1.8.14 Material design minimum temperature (T_R)

A characteristic minimum temperature of a material. It is used in design to select material with sufficient notch toughness to avoid brittle fracture and is the lowest temperature at which the material can be used at full design strength.

NOTE: The term 'material design minimum temperature' was denoted as 'MDMT' in previous editions of this Standard. The notation T_R has been adopted in this edition to allow for inclusion in various equations.

1.8.15 Maximum operating temperature (T_{max})

The highest metal temperature to which the vessel part under consideration is subjected under normal operation. It is determined by the technical requirements of the process (see Clause 3.2.2.4 for maximum service temperature for liquefied gas).

1.8.16 Maximum operating pressure

The highest pressure to which the vessel part under consideration is subjected under normal operation. It is determined by the technical requirements of the process (see Clause 3.2.1).

1.8.17 May

Indicates the existence of an option.

1.8.18 Minimum calculated thickness

The minimum thickness calculated according to the equations to resist loadings, before corrosion or other allowances are added.

1.8.19 Minimum operating temperature (T_{min})

The lowest temperature to which the vessel part under consideration is subjected during normal operation. It is determined by the technical requirements of the process or a lower temperature where specified by the purchaser.

NOTE: The term 'minimum operating temperature' was denoted as 'MOT' in previous editions of this Standard. The notation T_{min} has been adopted in this edition to allow for inclusion in various equations.

1.8.20 Minimum required thickness

The minimum thickness required, which is equal to the minimum calculated thickness plus corrosion and other allowances.

1.8.21 Nominal thickness

The nominal thickness of material selected as commercially available (and to which specified manufacturing tolerances are applicable).

1.8.22 Parties concerned

The purchaser, designer, fabricator, manufacturer, design verifier, inspection body, supplier, installer and owner as appropriate.

1.8.23 Pressure, calculation

The pressure (internal or external) used in conjunction with the design temperature to determine the minimum thickness or dimensions of the vessel part under consideration (see Clause 3.2.1).

1.8.24 Pressures

Unless otherwise noted, all pressures used in this Standard are gauge pressures or the difference in pressures on the opposite sides of the vessel part.

1.8.25 Pressure vessel

A vessel subject to internal or external pressure. It includes interconnected parts and components, valves, gauges and other fittings up to the first point of connection to connecting piping. It also includes fired heaters and gas cylinders, but excludes any vessel that falls within the definition of a boiler or pressure piping in AS/NZS 1200.

NOTES:

1 Gas cylinders are not covered by this Standard. It is intended that the above definition includes vessels, such as heat exchangers, evaporators, air receivers, steam type digesters, steam type sterilizers, autoclaves, reactors, calorifiers and pressure piping components, i.e. separators, strainers and the like. See Clause 1.3 for vessels specifically included and excluded.

2 Throughout this Standard 'pressure vessels' are referred to as 'vessels'.

1.8.26 Qualified welding procedure

A welding procedure satisfying the requirements of AS/NZS 3992.

1.8.27 Shall

Indicates that a statement is mandatory.

1.8.28 Should

Indicates a recommendation.

A2

1.8.29 Pressure piping

An assembly of pipes, pipe fittings, valves and piping accessories subject to internal pressure, external pressure, or both, and used to contain or convey fluid or to transmit fluid pressure. It includes distribution headers, bolting, gaskets, pipe supports and pressure-retaining accessories.

1.9 UNITS

Except where specifically noted, units used in the Standard are based on newtons, millimetres and degrees Celsius.

1.10 NOTATION

Symbols used in equations in this Standard are shown in Table 1.10.

Symbols not shown in Table 1.10 are defined in relation to the particular equations in which they occur.

1.11 PURCHASER AND MANUFACTURER

This Standard is written on the basis that the purchaser provides a suitable specification for the vessel, and the manufacturer performs all tasks required to construct the vessel according to the specification and to the requirements of this Standard. It is intended that, so long as the requirements of the Standard are met, the parties concerned can come to a documented agreement that certain tasks be performed by alternative parties.

NOTE: All parties should note that various laws, regulations, codes of practice and other requirements apply to pressure vessels in Australia.

Appendices E and F summarize the information required in various Clauses to be supplied by the purchaser and manufacturer, respectively.

1.12 REFERENCED DOCUMENTS

A list with titles of the documents referred to in this Standard, is given in Appendix R.

Where reference is made to a Standard by its number only, the reference applies to the current edition of the Standard, including amendments, unless otherwise agreed by the parties concerned. Where reference is made to a Standard by number, year and, where relevant, an amendment number, the reference applies to that specific document.

TABLE 1.10
MAIN SYMBOLS AND UNITS USED IN EQUATIONS

Description	Symbol	Units
Area	A	mm^2
Charpy–V rupture energy	C_v	J
Corrosion or erosion allowance	c	mm
Cycles–number of	N, n	—
Density	ρ	g/mm^3 or kg/m^3
Diameter	D, d	mm
Dimensions	English alphabet letters	—
Elongation at rupture	A_5	%
Force	F	N
Hardness	HB, HV	—
Joint factor (efficiency)	η	decimal
Length	L, l	mm
Linear expansion coefficient	α	µm/m°C
Mass	m	kg or tonne
Moment	M	Nm or Nmm
Plane angle	Any Greek letter	—
Poisson's ratio	υ	—
Pressure–design (or calculation)	P	MPa
Pressure–test	P_t	MPa
Radius	R, r	mm
Safety factor	S	—
Second moment of area	I	mm^4
Section modulus	Z	mm^3
Shear modulus	G	MPa
Strain	ε	%
Strength*–0.2% proof	$R_{p0.2}$	MPa

(*continued*)

TABLE 1.10 (*continued*)

Description	Symbol	Units
Strength*–0.2% proof – at temperature T	$R_{p0.2/T}$	MPa
Strength*–1% proof	$R_{p1.0}$	MPa
Strength*–1% proof – at temperature T	$R_{p1.0/T}$	MPa
Strength–creep rupture – for t hours at temperature T	S_{RT}	MPa
Strength–design	f	MPa
Strength*–tensile	R_m	MPa
Strength*–tensile–at temperature T	R_{mT}	MPa
Strength*–upper yield	R_{eH}	MPa
Strength*–yield	R_e	MPa
Strength*–yield–at temperature T	R_{eT}	MPa
Stress–normal	σ	MPa
Stress–shear	τ	MPa
Temperature	T	°C or K
Temperature–material design minimum	T_R	°C
Temperature–maximum allowable	T_{max}	°C
Temperature–minimum allowable	T_{min}	°C
Temperature–test	T_{test}	°C
Thickness–minimum calculated	t	mm
Thickness–minimum	t_{min}	mm
Thickness–nominal	t_n	mm
Time	t	s, min or h
Volume	V	mm^3, m^3 or L
Weight	W	N or kN
Young's modulus	E	MPa

* Specified minimum strengths

SECTION 2 MATERIALS

2.1 MATERIAL SPECIFICATIONS

2.1.1 General

Parent materials and components used in the construction of vessels shall comply with one of the following:

(a) Appropriate Standards listed in Tables B1, B2, 1.7B or Clause 2.2.

(b) Standards listed in AS 1228 or AS 4041.

(c) Material Standards published by ISO, ASTM (as listed in ASME BPV-VIII) or CEN for use in boilers, pressure vessels, or pressure piping.

(d) Other materials, in compliance with Clause 2.3.

Such materials shall be used within the limits given in the material or component Standard, and in this Standard.

NOTE: In some cases, material selection might be a matter for discussion between the parties concerned.

Piping and piping components used in pressure vessels shall have a construction suitable for the class of vessel.

NOTE: For example, if the joint efficiency of a piping grade is less than 1, it would be unsuitable for a Class 1 pressure vessel design.

Table 1.7B indicates the vessel Class limits for materials.

2.1.2 Grades

Only those material grades that are suitable for pressure containing parts and their integral attachments, for fabrication and for the conditions of operation for which the vessel is designed, shall be used.

Materials used in the manufacture of welded vessels shall be of satisfactory welding quality. Qualification of welding procedures in accordance with AS/NZS 3992 shall be considered as a minimum proof of satisfactory welding quality of the material. Materials used in the manufacture of brazed vessels shall be of satisfactory brazing quality. Qualification of brazing procedures in accordance with AS/NZS 3992 shall be considered as a minimum proof of satisfactory brazing quality of the material.

Steels of Groups A to E, inclusive (see Table 1.7B), used in the manufacture of welded pressure vessels that will be subjected to prolonged holding at temperature during postweld heat treatment(s) (e.g. exceeding 6 h total holding time) shall have representative test pieces subjected to the total simulated postweld heat treatment cycle(s). Such test pieces shall be subjected to mechanical testing requirements of the parent metal specification to ensure that any degradation in material properties from such heat treatment has not resulted in the material not meeting specification requirements.

Alloy steels may be selected for either creep or corrosion service. This will normally entail tempering temperatures at the bottom range for creep service and at the upper range for corrosion service. Such variations in tempering temperatures shall be taken into account in the selection of materials.

Plate materials that are used as a base in the manufacture of vessels with integrally clad plate or having applied corrosion-resistant linings, shall comply with the requirements of materials given in Table B1. Metal used for applied corrosion resistant lining may be any metallic material of weldable quality that is suitable for the intended service and acceptable to the purchaser.

Materials used for supporting lugs, saddles, skirts, baffles and similar non-pressure parts welded to vessels shall be of weldable quality and suitable in other respects for the intended service.

For Group F steels, see Clause 2.5.4.

2.2 STANDARD COMPONENTS AND INTEGRALLY CLAD METALS

Standard components, e.g. flanges, nozzles, pipe fittings, bolting and valves, and integrally clad metals used in vessel manufacture shall comply with the requirements of the following relevant Standards, except as provided in Clause 2.3.

(a) *Pipe fittings* ASME B16.9, ASME B16.11, AS/NZS 4331.

(b) *Pipe flanges* AS 2129, AS/NZS 4331, BS 3293, EN 1092, ASME B16.47, ASME B16.5.

(c) *Bolting* AS 1110 (series), AS 1111 (series), AS 1112 (series), AS/NZS 2451, AS/NZS 2465, AS 2528, AS B148, BS 2693.1, BS 4439, BS 4882, ASTM A 193, ASTM A 194, ASTM A 320.

(d) *Pipe threads* AS ISO 7.1, AS 1722.2, ASME B1.20.1, ASME B1.20.3, API Std 5B.

(e) *Valves* AS 1271, ASME B16.34.

(f) *Integrally clad plate* ASTM A 263, ASTM A 264, ASTM A 265, ASTM B 898.

NOTES:

1 These materials are required where design calculations are based on the total thickness, including cladding.

2 Bonding strength to be established in accordance with these specifications.

3 Where any part of the cladding thickness is specified as a corrosion allowance, such added thickness should be removed before tensile tests are carried out.

2.3 ALTERNATIVE MATERIAL AND COMPONENT SPECIFICATIONS

2.3.1 General

Where a material or component conforming to one of the specifications permitted by Clauses 2.1 or 2.2 is not available or desired, alternative materials and components may be used provided they comply with the requirements of AS/NZS 1200 for new or alternative materials.

Materials and components that are acceptable under ISO, British, American or European pressure equipment Standards are acceptable for use with this Standard (AS 1210).

2.3.2 Alternative product form

When there is no specification covering a particular product form of a wrought material for which there are specifications covering other product forms, such a product form may be used, provided the following conditions apply:

(a) The chemical, mechanical and physical properties, scope of testing, heat treatment requirements, and deoxidation or grain size requirements conform to specifications included in Table B1 or B2. The strength values for that specification, listed in Table B1 or B2, shall be used.

(b) The manufacturing procedures, tolerances, tests and marking are in accordance with the specification covering the same product form of a similar material.

(c) The two conditions in Items (a) and (b) are compatible in all respects, e.g. welding and testing requirements in Item (b) are appropriate for the material specified in Item (a).

(d) For welded pipe and tube made from plate, sheet or strip, without the addition of filler metal, 0.85 times the appropriate design strength, as listed in Table B1 or B2 or as derived in accordance with Appendix A, is used.

(e) Material manufacturer's test certificates reference the specifications used in producing the material.

2.3.3 Use of structural and similar quality steels for pressure parts

Structural and similar quality carbon and carbon-manganese steel plates and sheets, tubes, bars and sections not listed in Table B1, may only be used for pressure parts of Class 3 vessels, provided the following conditions are fulfilled:

(a) The specified minimum tensile strength of the steel is no greater than 460 MPa.

(b) The ladle analysis does not exceed the following:

Carbon.. 0.25%.

Phosphorus ....0.040%.

Sulfur ....0.040%.

Carbon equivalent based on: $C+\dfrac{Mn}{6}+\dfrac{Cr+Mo+V}{5}+\dfrac{Cu+Ni}{15}$ .... 0.45%.

(c) Test certificates (or equivalent) identifying the steel to a national Standard are provided and the steel is suitably marked or labelled.

(d) Plate used for flanges is not greater than 40 mm thick and the steel is not greater than 16 mm thick for tubes, sections and machined sockets and bosses, or 40 mm diameter for bars.

(e) Welded tube complies with a specification that requires hydrostatic testing of the tube. Regardless of the vessel classification, a maximum weld joint efficiency of 0.65 is to be used for welded tubes. Factor 0.85 in Clause 2.3.2(d) and factor 0.92 in Item (f) do not apply.

(f) The design strength for calculation purposes is determined in accordance with Appendix A and multiplied by a factor of 0.92.

(g) All weld preparations, openings, tubes, bars and sections shall be visually examined for evidence of lamination that may render the plate unacceptable (see 'Defects in materials' in AS 4458).

(h) The design temperature of the vessel is between 0°C and 250°C.

(i) The vessel is not used for applications with high risk of lamellar tearing or hydrogen blistering.

(j) If the steel is to be hot-formed above 650°C or normalized during fabrication, the specified properties of the material shall be verified by tests on a specimen subjected to a simulated heat treatment equivalent to that which the steel is to be subjected.

(k) Bars and sections that have been manufactured by a cold forming process are not acceptable unless an appropriate heat treatment is used, e.g. normalizing.

2.3.4 Specifically tested materials

The use of steel outside the limits of Clause 2.3.3 or other materials is permitted for pressure parts and attachments, provided—

(a) the material is shown by special tests to be equally suitable for the particular application as a similar material listed in Table B1; and

(b) agreement is given by the parties concerned.

These special tests may include chemical analysis, mechanical tests and non-destructive examination.

2.4 MATERIAL IDENTIFICATION

Material identification shall comply with the requirements of the material or component Standard and AS 4458.

2.5 LIMITS OF APPLICATION OF MATERIALS AND COMPONENTS

2.5.1 Maximum pressure limits

The maximum pressure for cast iron pressure parts should be in accordance with Notes 2 and 3 to Table B1(D).

Components shall be limited to the maximum pressures for which they are rated by the component specification and by the requirements in this Standard for the specific type of component.

2.5.2 Temperature limits

For low and high temperature limits, see Clauses 2.6 and 2.7, respectively.

2.5.3 Service limits

2.5.3.1 *Cast iron*

Grey and malleable cast irons and ductile cast irons*, with elongation less than 14% (on a gauge length of 5.65√area), shall not be used for vessels containing lethal or flammable fluids.

2.5.3.2 *Low melting point metals*

The low melting points of copper and aluminium and some of their alloys shall be taken into account where vessels will contain flammable fluid.

Materials for which no design strengths are given above 350°C in this Standard shall not be used for transportable vessels containing lethal substances, nor those containing flammable substances unless the vessel is insulated in accordance with Clause 3.26.

2.5.3.3 *Corrosion resistance*

In selecting material for vessels, consideration shall be given to the possibility of general or local wastage, corrosion, stress corrosion, corrosion fatigue, abrasion, erosion, and the like.

NOTE: For recommended corrosion practice, see Clause 2.9 and Appendix D.

2.5.4 Structural attachments and stiffening rings

Where pressure parts are constructed of Group F steel, all permanent structural attachments and stiffening rings welded directly to the pressure part shall be of nine percent nickel steel or an austenitic stainless steel which cannot be hardened by heat treatment. Where austenitic stainless steel is used for attachments, consideration shall be given to the greater coefficient of expansion of the austenitic stainless steel.

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* Alternative names for ductile cast iron are 'spheroidal graphite iron', 'SG iron' and 'nodular graphite iron'.

2.6 MATERIAL FOR LOW TEMPERATURE SERVICE

2.6.1 General

Materials and components for pressure parts (and for non-pressure parts welded directly to pressure parts) for low temperature service, or where it is required to guard against brittle fracture, shall comply with the appropriate requirements of this Clause (2.6). These requirements do not apply to non-pressure parts such as internal baffles, trays, supports, and the like where these are not attached to a pressure part by welding and are not otherwise an integral part of a pressure part.

The following methods may be used to establish requirements to guard against low temperature brittle fracture:

(a) Method 1: Technical requirements in Clauses 2.6.2 to 2.6.5, developed from testing, operating experience and fracture mechanics principles using Charpy V values to characterize metal toughness.

(b) Method 2: Fracture mechanics analysis in accordance with Clause 2.6.6, by agreement of the parties concerned.

See Clause 3.21.5 for requirements for steel bolting for low temperatures.

Unless otherwise noted, metal temperatures are mean values.

2.6.2 Selection of material

2.6.2.1 *General*

To select suitable material for each part of the vessel, the following procedure may be used:

(a) For carbon, carbon-manganese and low alloy wrought or cast steels but excluding bolting, either see Clauses 2.6.2.2, 2.6.2.3 and 2.6.2.4, or determine the following:

  (i) The minimum operating temperature (T_{min}) for the part from Clause 2.6.3.1;

  (ii) The required material design minimum temperature* (T_R) from Clause 2.6.3.2;

  (iii) The material reference thickness (t_R) from Clauses 2.6.4;

  (iv) The impact test temperature (T_{cv}); and

  (v) Select a suitable steel from Table 2.6.2.

(b) For metals other than carbon, carbon-manganese and low alloy wrought or cast steel and excluding steel bolting—

  (i) determine the T_{min} for the part in accordance with Clauses 2.6.3.1; and

  (ii) from Table 2.6.3, select the permitted material (and any necessary impact tests) having a required T_R equal to or less than T_{min}.

  NOTE: Where reference is made in Table 2.6.3 to Figure 2.6.2(A) or Figure 2.6.2(B), see Item (a) for guidance.

(c) For non-metallic materials, see Clause 2.6.7.

The sequence in items (a) and (b) may be changed as appropriate to determine T_{min}, T_R or t_R.

2.6.2.2 *Thin-walled carbon and carbon-manganese steel tubes (seamless and welded)*

Heat exchanger tubes of carbon and carbon-manganese and low alloy steels with less than 0.25% carbon and a specified minimum tensile strength of less than 450 MPa, may be used at T_{min} as shown in Table 2.6.2.2 provided—

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* The 'material design minimum temperature' was denoted as 'MDMT' in previous editions of this Standard. The notation T_R has been adopted in this edition to allow for inclusion in various equations.

(a) the tubes are used in heat exchangers of the floating head type;

(b) the tubes used in U-tube type heat exchangers are heat-treated after cold bending where required by AS 4458; or

(c) for fixed tubeplate type heat exchangers, it is demonstrated that tensile stresses in the tubes due to differential thermal expansion are low, e.g. where spiral-wound tubes or expansion bellows are used, or calculated tensile stresses are less than 50 MPa.

TABLE 2.6.2.2

MATERIAL DESIGN MINIMUM TEMPERATURE FOR HEAT EXCHANGER TUBES

Thickness	Tube to tubesheet attachment method		
	As welded	Welded and PWHT	Unwelded
mm	°C	°C	°C
10	−15	−30	−70
8	−20	−35	−75
6	−25	−40	−80
4	−40	−55	−95
2	−55	−70	−110

2.6.2.3 *Thin materials*

Materials having a thickness insufficient to obtain a 2.5 mm Charpy V-notch specimen may be used at a temperature no less than that permitted for non-impact tested material of similar type, or as provided in Clause 2.6.2.2 and Clause 2.6.2.4, or as established by tests agreed by the parties concerned.

2.6.2.4 *Metals not requiring impact testing*

Impact testing of metals is not required for the following:

(a) Steels covered by curves T_{cv} = 0°, 20° and 30° of Figures 2.6.2(A) or (B), for the appropriate T_R and service conditions.

(b) Steels covered by curves T_{cv} = −20° and −40°, for T_R 10°C above that from Figures 2.6.2(A) and (B).

(c) Carbon and carbon-manganese steels Groups A1, A2 and A3 with specified carbon content above 0.25% covered by curves T_{cv} = 0°, 20° and 40°, for T_R 10°C above that from Figures 2.6.2(A) and (B).

(d) Steels covered by curves T_{cv} = 20°, 40° and 50° thinner than 3.0 mm, provided they are used at design metal temperatures not colder than −50°C.

See also Table 2.6.2.2 and Note 5 to Table 2.6.2.

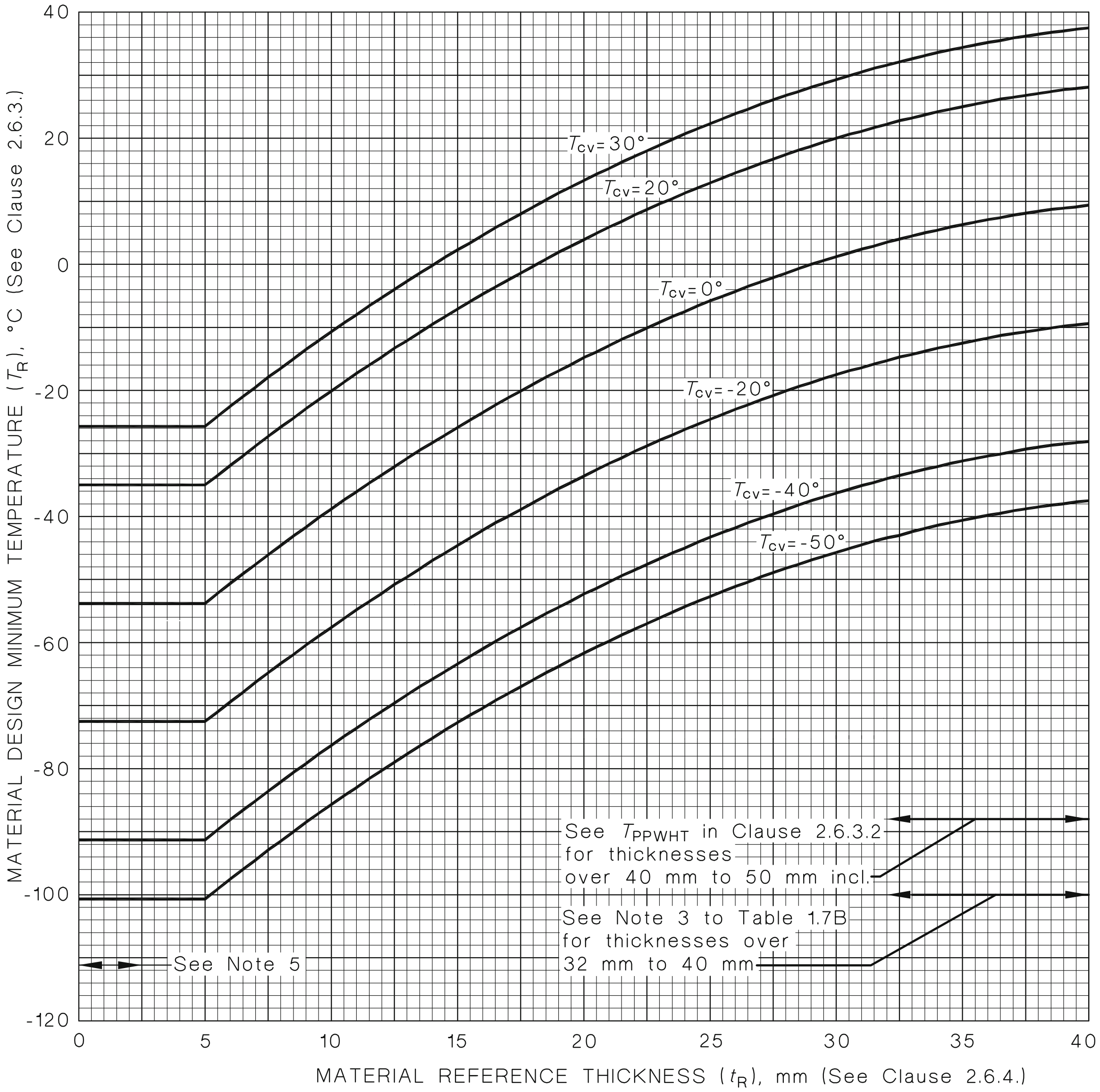
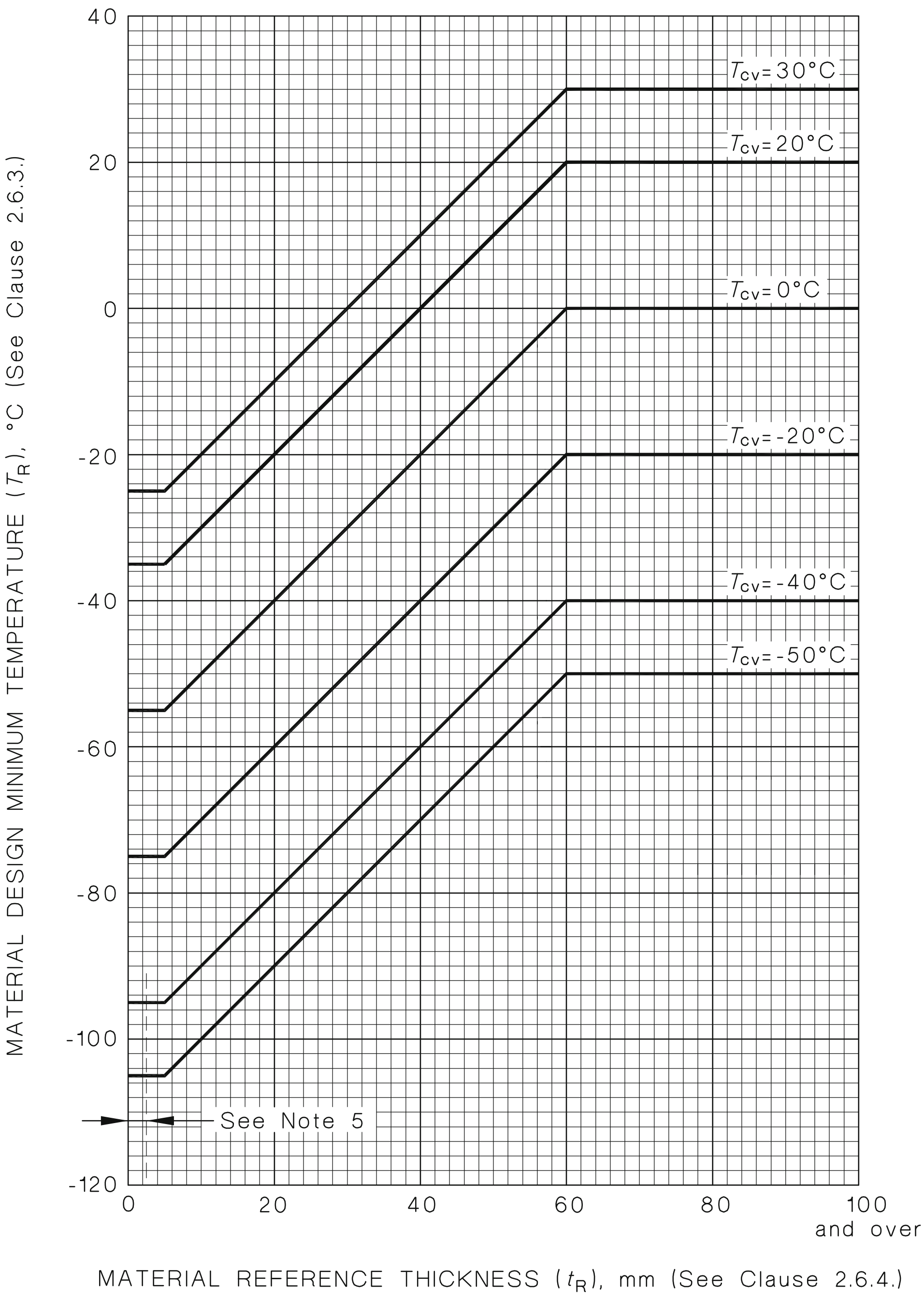


FIGURE 2.6.2(A) WROUGHT CARBON, CARBON-MANGANESE AND LOW ALLOY STEELS—SELECTION OF MATERIAL FOR LOW TEMPERATURE SERVICE —AS WELDED



*Notes to this figure follow Table 2.6.2.

FIGURE 2.6.2(B) WROUGHT CARBON, CARBON-MANGANESE AND LOW ALLOY STEELS—SELECTION OF MATERIAL FOR LOW TEMPERATURE SERVICE—POSTWELD HEAT TREATED AND UNWELDED

TABLE 2.6.2

BASIS FOR CURVES ON FIGURES 2.6.2(A) AND 2.6.2(B)

Curve T_{cv} °C (Notes 5 and 6)	Standard impact test value (*J*) Specified minimum yield strength, MPa			Steels represented by curves	
	$R_e \leq 310$	$310 < R_e \leq 360$	$360 < R_e \leq 450$	Type (Note 8)	AS 1548 Grades
30 No test (Note 7)	—	—	—	All, C < 0.35%	All
20 No test (Note 7)	—	—	—	All, C≤0.25%	All
0 (Note 1)	27	31	40 (Note 3)	All, C <0.25%	All
0 No test (Note 7)	—	—	(Note 4)	Fine grained C-Mn steel	PT430 L0 to L40 PT460 L0 to L50 PT490 All designations PT540 All designations
–10 (Note 1)	27	31	40 (Note 3)	Fine grained C-Mn steel (Note 2)	PT430 L0 to L40 PT460 L0 to L50 PT490 All designations PT540 All designations
–20 (Note 1)	27	31	40 (Note 3)	Fine grained C-Mn steel (Note 2)	PT430 L0 to L40 PT460 L0 to L50 PT490 All designations PT540 All designations
–40 (Note 1)	27	31	40 (Note 3)	Fine grained C-Mn steel (Note 2)	PT430 L20, L40 PT460 L20-L50 PT490 L20-L50 PT540 L40, L50
–50 (Note 1)	27	31	40 (Note 3)	Fine grained C-Mn steel (Note 2)	PT430 L0 to L40 PT460 L0 to L50 PT490 All designations PT540 All designations

NOTES TO FIGURES 2.6.2(A), 2.6.2(B), AND TABLE 2.6.2:

1 Tested by steelmaker or manufacturer.

2 Steels produced to fine grained practice, i.e.—

(a) normalized steel where the actual Mn% divided by the actual C% ≥4;

(b) controlled rolled;

(c) thermo-mechanically controlled rolled;

(d) grain refining elements added, e.g. aluminium or titanium (or both) 0.01% minimum. Examples are AS 1548 and AS/NZS 1594 grade HU 300/1; or

(e) ferritic grain size is 6 or finer, when tested to ISO 643.

3 For steel with impact test values equal to or greater than 27 J and less than 40 J, material design minimum temperatures 10°C above the curve will apply. Where a Standard does not specify the temperature corresponding to 27 J (or 31 J or 40 J) Charpy V energy, the value specified may be converted to the 27 J (or 31 J or 40 J) temperature on the basis of 1.5 J/°C. This conversion shall be permitted in the range of Charpy V energy 18 J to 50 J. For example, AS 1548 Grade PT460 giving 47 J at −20°C may be regarded as equivalent to 27 J at −33°C.

4 Applicable only to steel with specified minimum tensile strength equal to or less than 490 MPa.

5 Impact tests are not required for material thinner than 3 mm or where it is impracticable to obtain a 10 mm × 2.5 mm specimen. (See also Clause 2.6.2.3). Material specifications might not require impact tests on Charpy specimens smaller than 10 mm × 5 mm without special negotiation and thus impact tested material thinner than 7 mm might not be readily available. Extrapolation of curves from 5 mm to 2.5 mm, where agreed between the parties concerned, should be supported by suitable testing.

6 Values at intermediate test temperatures may be obtained by linear interpolation.

7 Where impact testing is not required and the vessel is classified as a hazard level A or B to AS 4343 and ferritic steels are purchased from an unproven supplier, a Charpy-V-impact test should be performed on at least one item of product to satisfy curves T_{cv} = 20°, 30° and 40° of Figure 2.6.2(B) at these test temperatures.

8 The maximum permitted carbon content (C) by cast analysis is 0.25 % for curves T_{cv} = −20°, −40° and −50°. This limit may require restriction of the normally specified carbon content in some steel specifications permitted by this Standard.

TABLE 2.6.3

MATERIAL DESIGN MINIMUM TEMPERATURE (T_R)

Material		**Typical specification or nominal composition**		**Material design minimum temperature (T_R), °C (Note 1)**	
Steel group (Note 3)	**General type**	**Standard of specification**	**Grade or type**	**Not impact tested**	**Impact tested (Note 4)**
CARBON AND CARBON-MANGANESE STEEL (all forms excluding weld metal and bolting)					
A1	C, C-Mn (low strength)	AS 1548	PT430, PT460	See Clauses 2.6.2 and 2.6.3.2	See Clauses 2.6.2 and 2.6.3.2 and Note 5
A2	C, C-Mn (medium strength)	AS 1548	PT490, PT490		
A3	C, C-Mn (high-yield strength)	AS 1548 AS/NZS 1594	PT540 XF 400, XF 500		
A4	C, C-Mn (high-yield strength) Quenched and tempered, with or without boron	 JIS-G3115	ASME Group P1.4 EN Group 1.3 SPV 490		
LOW ALLOY STEEL (all forms, excluding weld metal and bolting) (Note 2)					
B	Cr or Mo < ¾	—	C - ½Mo, ½Cr -½Mo	T_R for curve T_{cv} = 20° in Figure 2.6.2 (A) or (B) as appropriate but not lower than 0°C	Use the appropriate curve in Figures 2.6.2(A) and (B)
C	¾ ≤ Total alloy ≤ 3	—	1Cr -½Mo		
D1	Vanadium	—	½Cr -½Mo -¼V		
D2	3 ≤ Total alloy ≤ 10		2¼Cr-1Mo		
E	1½-5 Ni	ASTM A 203	D E	–30 or T_R for curve T_{cv} = 0° in Figure 2.6.2 (A) or (B), whichever is less	Test temperature giving C_v ≥ 27 J
F	9 Ni	ASTM A 353	A, B, C, D, E, F, J, P	Not permitted	Test temperature giving C_v ≥ 20 J
G	Quenched and tempered	ASTM A 517, AS 3597	700 PV		Test temperature giving lateral expansion ≥0.38 mm for each specimen (Note 9) and NDTT (Note 6)

(*continued*)

TABLE 2.6.3 (*continued*)

Material		Typical specification or nominal composition		Material design minimum temperature (T_R), °C (Note 1)	
Steel group (Note 3)	**General type**	**Standard of specification**	**Grade or type**	**Not impact tested**	**Impact tested (Note 4)**
HIGH ALLOY STEEL (all forms, excluding castings, weld metal and bolting)					
H	Martensitic chromium Types 12 Cr and 15 Cr	ASTM A 240	410, 429	T_R as for curve T_{cv} = 20° in Figure 2.6.2 (A) or (B) as appropriate but not lower than −30°	
J	Ferritic high chromium types 12 Cr-Al or 12 Cr low C	ASTM A 240	405, 410S		
K	Austenitic chromium nickel types (only plate specifications shown) (See Note 10):				
	18 Cr-8 Ni	ASTM A 240	304	−255	
	18 Cr-8 Ni (Low C)	ASTM A 240	304L	−255	
	18 Cr-8 Ni-Nb	ASTM A 240	347	−255	
	18 Cr-10 Ni-Ti	ASTM A 240	321	−200 (Note 7)	Test temperature giving lateral expansion ≥ 0.38 mm for each specimen or Test temperature giving C_v ≥ 27 J where R_e ≤ 310 MPa; 40J where 310 MPa< R_e< 650 MPa
	18 Cr-10 Ni-2 Mo	ASTM A 240	316	−200	
	18 Cr-10 Ni-2 Mo(Low C)	ASTM A 240	316L	−200	
	19 Cr-13 Ni-3 Mo	ASTM A 240	317	−200	
	25 Cr-20 Ni	ASTM A 240	310S	−200	
	Selected Austenitic types postweld heat treated below 900°C	ASTM A 240	309, 310, 316, 309Cb, 310Cb, 316Cb	Not permitted	
	Any type with C > 0.10%	ASTM A 240	302	−30	

(*continued*)

TABLE 2.6.3 (*continued*)

Material		Typical specification or nominal composition		Material design minimum temperature, °C (Note 1)	
Steel group (Note 3)	General type	Standard of specification	Grade or type	Not impact tested	Impact tested (Note 4)
L	High chromium	ASTM A 240	442,446	Not permitted	
M	Ferritic-austenitic chromium nickel	ASTM A 789	S31803	T_R as for curve T_{cv} = 20° in Figure 2.6.2 (A) or (B) as appropriate but not lower than −30. For unwelded material ≤5 mm thickness, the T_R ≥−50°C.	
HIGH ALLOY STEEL (Castings)					
All types (H to M)		—	—	As for Group H steels	Test temperature giving C_v ≥ 20 J
CAST IRONS					
	Grey iron	AS 1830	T-150 to T-400	−30	Note 8
	Spheroidal graphite	AS 1831	500-7 and 400-12		
	Malleable iron	AS 1831	370-17	T_R as for curve T_{cv} = 20° in Figure 2.6.2 (B) but not lower than −30°	Test temperature giving C_v ≥ 20 J
	Austenitic iron	AS 1832	All whiteheart and blackheart	−30	Note 8
	Austenitic iron	AS 1833	All spheroidal graphite	−30	Test temperature giving C_v ≥ 20 J
NON-FERROUS METALS					
All types except Titanium and its alloys		See Tables B1, (D), (E), (F) and (H)		No Limit	Impact test not required
Titanium and its alloys		ASTM B 265		−60	Test temperature giving C_v ≥ 20 J

LEGEND:

C_v = Charpy V impact test values; R_e = specified minimum yield strength

NOTES TO TABLE 2.6.3:

1 See Clause 2.6.3.2 for adjustments permitted or required.

2 Low alloy steels not listed in or not equivalent to those in the table shall meet the requirements specified for Group B steels.

3 For steel groups, see Table 1.7 and AS/NZS 3992.

4 Where Charpy V energy values are quoted, the values are the minimum average values for each set of three 10 mm × 10 mm specimens.

5 For variations permitted for different energy values and test temperatures, see Note 3 to Figures 2.6.2(A) and 2.6.2(B).

6 In addition to Charpy V impact test, dropweight tests are required for—

(a) Group F steels ≥ 16 mm thick for use at minimum operating temperature (T_{min}) below −170°C; and

(b) Group G steels ≥ 16 mm thick for use at T_{min} below −30°C.

7 Impact testing for these high alloy steels is not required below the temperature listed when the calculated primary general membrane stress used for determining thickness does not exceed 50 MPa.

8 These cast irons may be used below −30°C by agreement of the parties concerned, on the basis of suitable testing or successful past experience.

9 For Group F and Group G steels, the maximum test temperature is 0°C.

10 For austenitic stainless steels, the T_R listed here applies to material in the solution-annealed condition only.

2.6.2.5 *Use of fracture mechanics*

Materials may be used at temperatures lower than otherwise required by Clauses 2.6.2 to 2.6.5, provided that suitable fracture mechanics analysis (see Clause 2.6.6) and tests justifying the lower temperatures are carried out.

2.6.2.6 *Welded material*

Where materials are welded, the following shall apply:

(a) The heat affected zones and weld metal of each weld shall meet the temperature limits and impact requirements for the component with the higher T_R, as specified in this Standard and in AS/NZS 3992.

(b) The weld procedure shall be qualified according to AS/NZS 3992.

(c) Weld production test plates shall be prepared in compliance with this Standard (see Clause 5.2.2) and AS/NZS 3992.

2.6.3 Minimum temperatures

2.6.3.1 *Minimum operating temperature (T_{min}) **

T_{min} shall be the lowest metal temperature of the part under consideration during normal operation including normal process fluctuations and during properly conducted start-up and shutdown. T_{min} shall be the lowest of the following:

(a) For vessels thermally insulated externally—the minimum temperature of contacting contents.

(b) For vessels not thermally insulated—the lower of—

(i) the lowest one day mean ambient temperature (LODMAT)† plus 10°C, where the metal can be subjected to this temperature while the shell is under pressure; and

---

* Minimum operating temperature was denoted as 'MOT' in previous edition of this Standard. The notation T_{min} has been adopted to allow its inclusion in various equations in this edition of this Standard.

† See Appendix K.

(ii) the minimum temperature of the contacting contents except that for Groups A1, A2, A3, A4, B, C, D1, D2, and G steels in vessels containing fluids at temperatures governed by atmospheric conditions only and whose vapour pressure reduces with decreasing temperature, the temperature corresponding to the vapour pressure obtained by dividing the vessel design pressure by 2.5 may be used.

(c) If there is evidence to show that because of radiation, adiabatic expansion or other effects, the above will not provide a reliable estimate of temperature, the method to be used in assessing the temperature shall be agreed. Allowance shall be made for any sub-cooling during pressure reduction (see Clause 2.6.3.2).

(d) A lower temperature than that determined from Items (a), (b) or (c) where so specified by the purchaser or an application Standard.

2.6.3.2 *Material design minimum temperature (T_R) for Group A to E steels*

For Group A to E steels, the required T_R (in degrees Celsius) shall be calculated using the following equation:

A1

$$T_R = T_{min} + T_S - T_L + T_{PPWHT} - T_{SHOCK} - T_C - T_{STRAIN} - T_H \quad \ldots 2.6.3$$

where

T_{min} = minimum operating temperature, in degrees Celsius, from Clause 2.6.3.1

T_S = temperature adjustment for stress

= 0°C for post weld heat treated (Note 1) and k_s >75%;

= 10°C for post weld heat treated (Note 1) and 50% <k_s ≤75%;

= 25°C for post weld heat treated (Note 1) and k_s ≤50%;

= 50°C for post weld heat treated (Note 1) and membrane stress ≤50 MPa (Note 2);

= 0°C for as-welded condition, membrane stress >50 MPa (Note 2) and t_R <30 mm; or

= 40°C for as-welded condition, and membrane stress ≤50 MPa (Note 2) and t_R <30 mm.

NOTES:

1 Also applies to non-welded parts and vessels where nozzles and non-temporary attachments are post weld heat treated, before assembly into the vessel with butt welds which are not subsequently post weld heat treated.

2 This membrane stress must take account of internal and external pressure and dead weight. For walls and pipes of heat exchangers the restraint of free end displacement of the heat exchanger pipes should also be taken into account.

k_s = ratio of pressure induced principal membrane stress and design strength for Class 1H vessels, *f*.

f = highest design stress for material under Class 1H, in megapascals

NOTE: The Class 1H value is to be used here for all Classes.

η = weld efficiency (for castings this shall be 0.8 max)

σ_m = calculated primary membrane stress in the corroded condition, in megapascals

T_L	=	15°C	for lethal vessel contents; or
		0°C	otherwise.
T_{PPWHT}	=	15°C	for partially post-weld heat treated parts where plates containing nozzles, supports or other welded attachments are postweld heat-treated before they are butt-welded to the shell, but the main joints are not postweld heat-treated, and where the minimum distance from the edge of the welds of the attachments to the main welded joints is not less than the lesser of $3t_{shell}$ and 150 mm; or
		0°C	otherwise.
T_{SHOCK}	=	15°C	for transportable vessels or other vessels subject to similar shock loadings; or
		0°C	otherwise.
T_C	=	10°C	for Class 2H vessels containing liquefied gas; or
		0°C	otherwise.
T_{STRAIN}	=	1°C	for each 1% cold forming strain greater than 5% for any part that is not subsequently heat treated
T_H	=	10°C	for vessels not subject to a full pressure test; or
		0°C	otherwise.

The calculated stresses shall take into account all loadings, such as internal and external pressures, thermal stress and external loads arising from connecting pipes. Where such a vessel will also be subject to a higher pressure at higher temperature, e.g. in a refrigeration system with liquefied gas, the material and design shall be suitable for all expected combinations of operating pressures and temperatures.

Where the vessel might be subject to cooling due to a pressure drop followed by a significant pressure increase while still cold, it is recommended the required T_R be equal to the atmospheric boiling point, for a liquefied gas, or the estimated minimum metal temperature, for a permanent gas.

2.6.3.3 *T_R for metals other than Group A to E steels*

For metals other than Group A to E steels, the required T_R shall be as specified in Clause 2.6.2.1.

2.6.4 Material reference thickness

The reference thickness (t_R) used in applying Figures 2.6.2(A) and 2.6.2(B) shall be
A2 determined from Table 2.6.4 and as follows:

(a) *Butt welded shell components* The reference thickness of each component shall be taken as the actual thickness of the component under consideration at the edge of the weld preparation.

(b) *Weld neck flanges, plate and slip-on (or hubbed) flanges, flat tubeplates and flat ends* The reference thickness shall be the greater of one-quarter the actual thickness of the flange, tubeplate or flat end, or the thickness of the nozzle or shell attached thereto.

If the distance from the flange, tubeplate or flat end to the butt weld is not less than four times the thickness of the butt weld, the reference thickness for the as-welded condition shall be the thickness at the edge of the weld preparation.

The reference thickness of tubeplates having tubes attached by welding shall be taken as not less than tube thickness.

NOTE: Where the shell to tubeplate joint is stress relieved but the tube/tubeplate joint is as-welded, this may affect the selection of materials for the tubeplate.

(c) *Nozzles and compensating plates* The reference thickness of each welded component shall be determined separately by considering only the actual thickness of that component. Where butt-welded inserts are used, the reference thickness shall correspond to the thickness at the edge of the weld preparation.

(d) *Tubes* The reference thickness shall be that of the actual thickness of the tube.

(e) *Attachments* Attachments welded directly to a pressure component shall be regarded as part of the pressure component, and the reference thickness shall be the thickness
A2 as shown in Table 2.6.4. Intermediate attachments (see Table 2.6.4, Item (g) Attachments) shall be employed where it is required to attach non-critical components to the shell.

(f) *Welds* The reference thickness for welds shall be the following:

(i) For welds in shells or nozzles, or attachments directly welded to shells or nozzles—the maximum thickness of the parts joined.

(ii) For welds in flanges, flat ends and flat tube plates—the minimum thickness of the parts joined.

(g) *Castings* The reference thickness shall be the maximum nominal thickness.

(h) *Unwelded components* Unwelded components shall be taken as postweld heat treated and the reference thickness shall be taken as one-quarter of the thickness of the item.

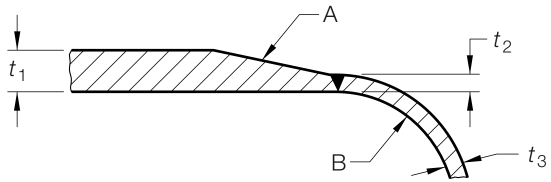
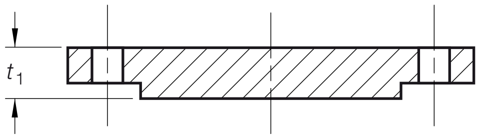
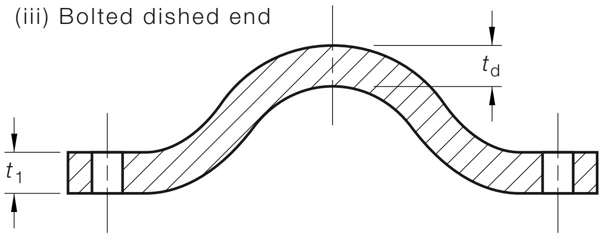
(i) The thickness above used as a basis for the reference thickness shall be the nominal thickness and shall be not less than the minimum calculated thickness plus corrosion and other allowances.

The reference thickness (t_R) and condition used to determine T_R for a component shall be the thickness and condition giving the warmest T_R.

NOTE: t_R is approximately equivalent to the thickness of a cylindrical shell with an assumed longitudinal weld defect, and with residual and applied stresses.

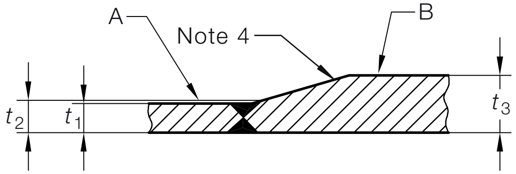
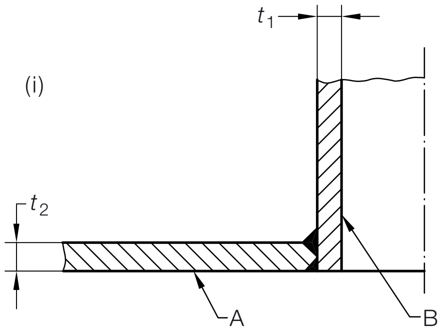
TABLE 2.6.4

DETERMINATION OF MATERIAL REFERENCE THICKNESS (t_R)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(a) Unwelded components (i) Seamless cylinder 	AW	Max of t_2 and $t_1/4$ in Figure 2.6.2(B)	t_2	t_3	—
	PWHT	Max of t_2 and $t_1/4$	t_2	t_3	—
(ii) Seamless flat plate 	Treat as PWHT	$t_1/4$	—	—	—
(iii) Bolted dished end 	Treat as PWHT	Max. of ($t_1/4$) and $t_d/4$	—	—	—

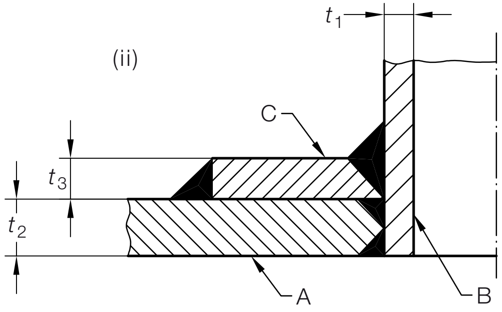
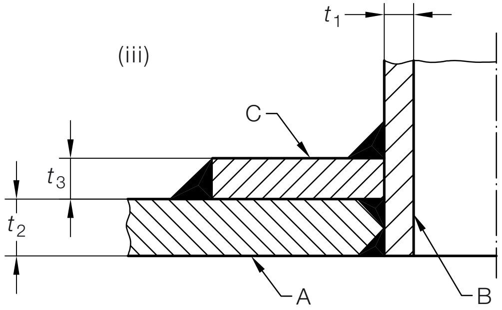
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(b) Butt-welded shell components – circumferential and longitudinal welds  	AW	t_1	t_2	Max of t_2 and $t_3/4$ in Figure 2.6.2(B)	—
	PWHT	t_1	t_2	Max of t_2 and $t_3/4$	—
(c) Nozzles (i)  	AW	t_2	t_2	t_1	—
	PWHT	t_2	t_2	t_1	—

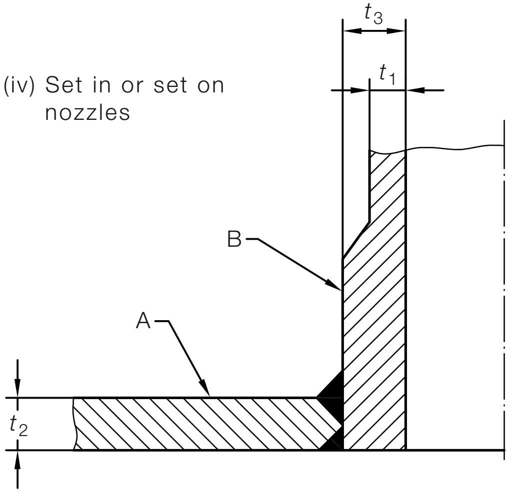
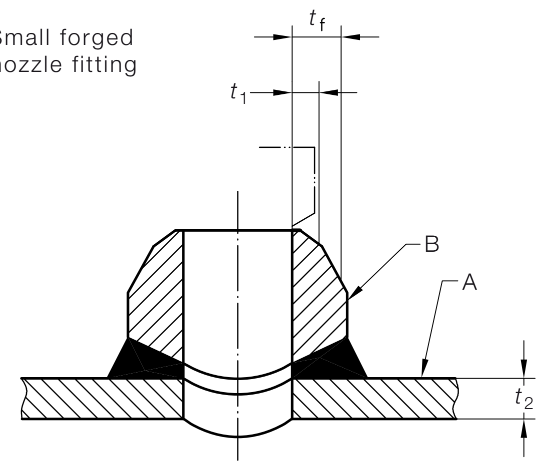
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(ii) 	AW	t_2	Max. of t_2 and t_3	t_1	t_3
	PWHT	t_2	Max. of t_2 and t_3	t_1	t_3
(iii) 	AW	t_2	Max. of t_2 and t_3	t_1	t_3
	PWHT	t_2	Max. of t_2 and t_3	t_1	t_3

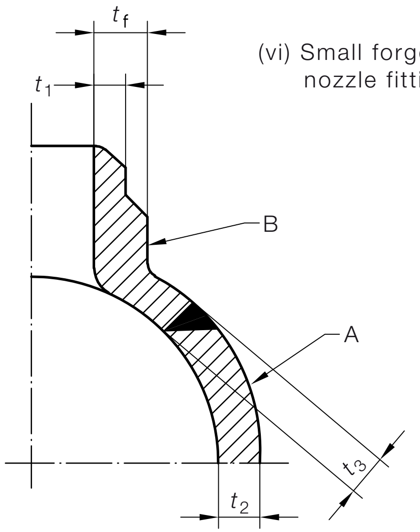
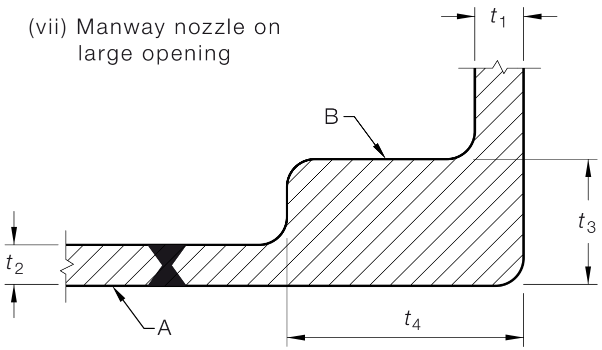
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(iv) Set in or set on nozzles 	AW	t_2	Max. of t_2 and t_3	t_3	—
	PWHT	t_2	Max. of t_2 and t_3	t_3	—
(v) Small forged nozzle fitting 	AW	t_2	t_2	Max. of t_1 and $t_f/4$	—
	PWHT	t_2	t_2	Max. of t_1 and $t_f/4$. If necessary, check t_1 in Figure 2.6.2(A).	—

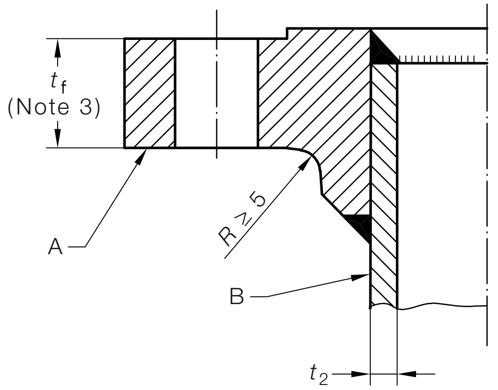
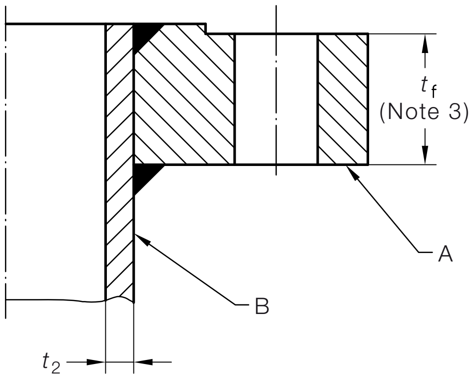
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(vi) Small forged nozzle fitting 	AW	t_2	t_3	Max. of t_3 and $t_f/4$	—
	PWHT	t_2	t_3	Max. of t_3 and $t_f/4$. If necessary, check t_1 in Figure 2.6.2(A).	—
(vii) Manway nozzle on large opening 	AW	t_2	t_2	Max of t_1; t_2; and the lesser of $t_3/4$ and $t_4/4$ in Figure 2.6.2(B)	—
	PWHT	t_2	t_2	Max of t_1; t_2; and the lesser of $t_3/4$ and $t_4/4$	—

(*continued*)

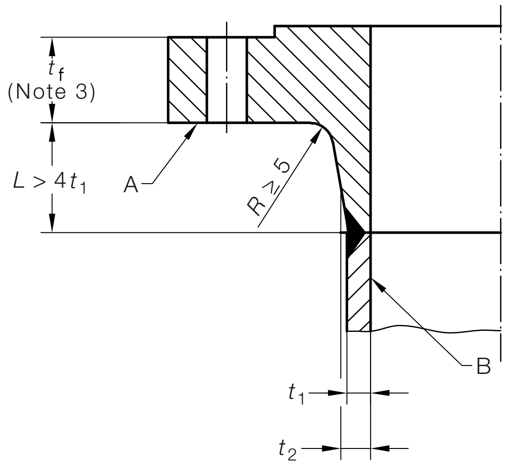
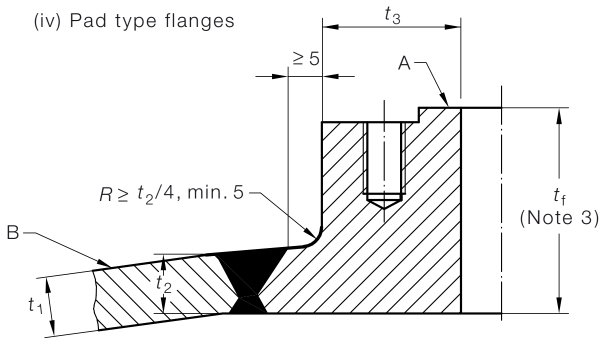
TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	**Weld**	**Part B**	**Part C**
(d) Flanges (i) Slip-on flange  	AW	$t_f/4$	t_2 for both welds	t_2	—
	PWHT	$t_f/4$	t_2 for both welds	t_2	—
(ii) Plate flange  	AW	$t_f/4$	t_2 for both welds	t_2	—
	PWHT	$t_f/4$	t_2 for both welds	t_2	—

(*continued*)

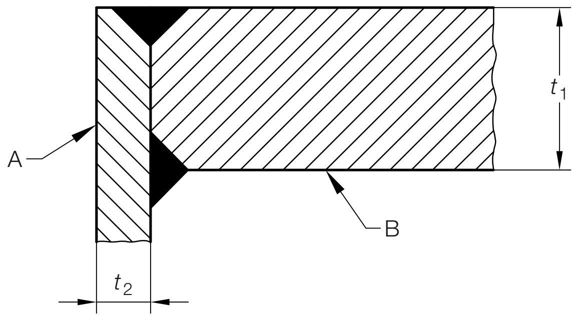
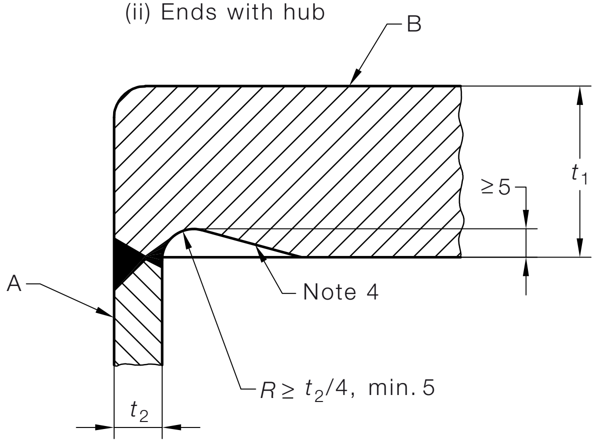
TABLE 2.6.4 (*continued*)

A2

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(iii) Forged or cast welding neck flanges 	AW	Max. of t_2 in Figure 2.6.2(A) and $t_f/4$ in Figure 2.6.2(B), whichever is the more onerous	t_2	t_1	—
	PWHT	Max. of t_2 and $t_f/4$	t_2	t_1	—
(iv) Pad type flanges 	AW	Max. of t_2, and the lesser of $t_3/4$ and $t_f/4$ in Figure 2.6.2(A)	t_2	t_1	—
	PWHT	Max. of t_2, and the lesser of $t_3/4$ and $t_f/4$	t_2	t_1	—

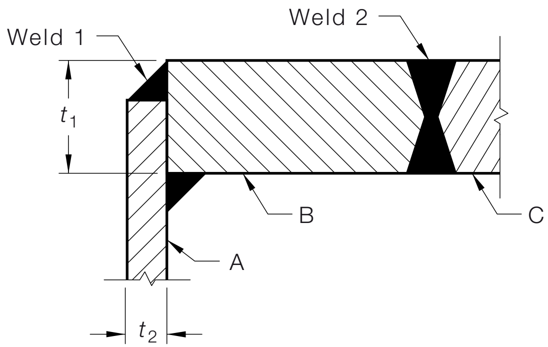
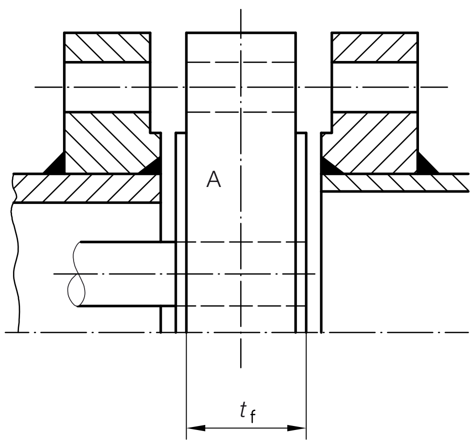
(*continued*)

TABLE 2.6.4 (*continued*)

<table>
<tr><th rowspan="2">Construction details</th><th rowspan="2">Condition (Note 1)</th><th colspan="4">Reference thickness, t_R (Note 2)</th></tr>
<tr><th>Part A</th><th>Weld</th><th>Part B</th><th>Part C</th></tr>
<tr><td rowspan="2">(e) Flat ends
(i) Flat ends
</td><td>AW</td><td>t_1</td><td>t_2</td><td>Max. of $t_1/4$ and t_2</td><td>—</td></tr>
<tr><td>PWHT</td><td>t_1</td><td>t_2</td><td>Max. of $t_1/4$ and t_2</td><td>—</td></tr>
<tr><td rowspan="2">(ii) Ends with hub
</td><td>AW</td><td>t_2</td><td>t_2</td><td>Max. of t_2, and $t_1/4$ in Figure 2.6.2(B)</td><td>—</td></tr>
<tr><td>PWHT</td><td>t_2</td><td>t_2</td><td>Max. of $t_1/4$ and t_2</td><td>—</td></tr>
</table>

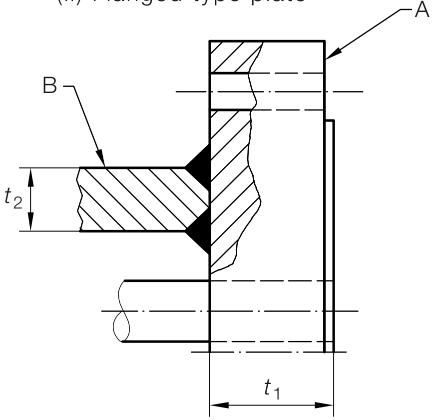
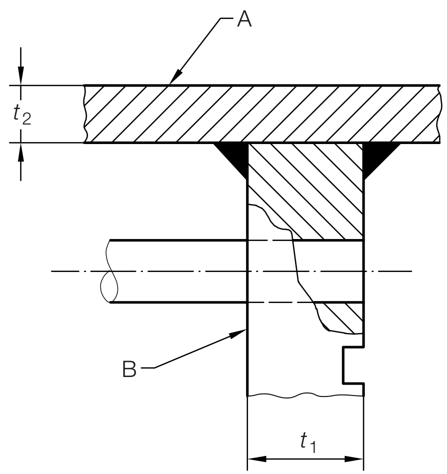
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	**Weld**	**Part B**	**Part C**
(iii) Weld in flat end 	AW	t_2	Weld 1: t_2 Weld 2: $t_1/4$	Max. of $(t_1/4)$ and t_2	Max. of $(t_1/4)$ and t_2
	PWHT	t_2	Weld 1: t_2 Weld 2: $t_1/4$	Max. of $(t_1/4)$ and t_2	Max. of $(t_1/4)$ and t_2
(f) Tube plates (i) 	Treat as PWHT	$t_f/4$	—	—	—

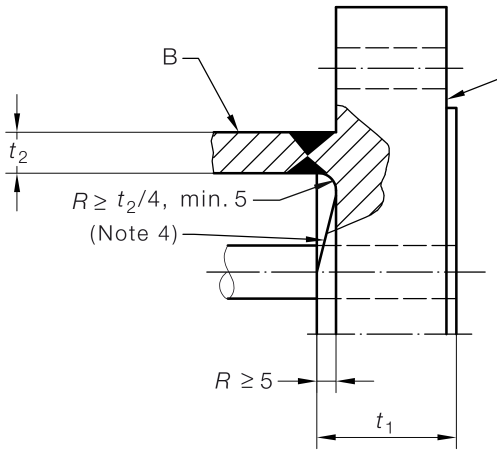
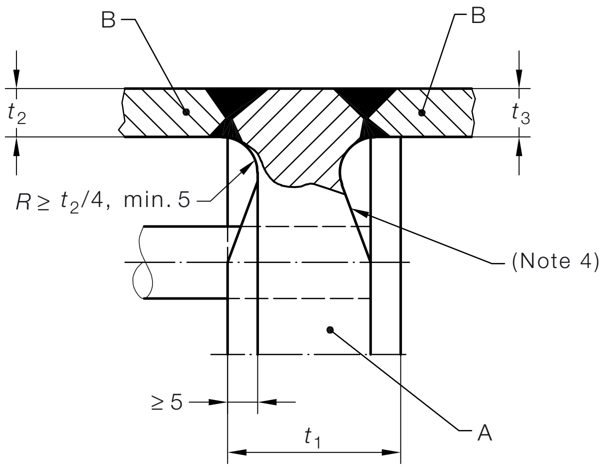
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	**Weld**	**Part B**	**Part C**
(ii) Flanged type plate 	AW	Max of (t_1/4) and t_2	t_2	t_2	—
	PWHT	Max. of (t_1/4) and t_2	t_2	t_2	—
(iii) Welded into shell/channel 	AW	Max. of (t_1/4) and t_2	t_2	t_2	—
	PWHT	Max. of (t_1/4) and t_2	t_2	t_2	—

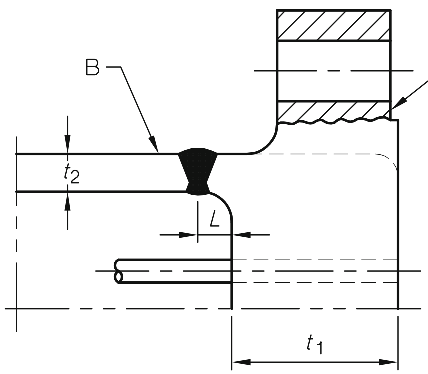
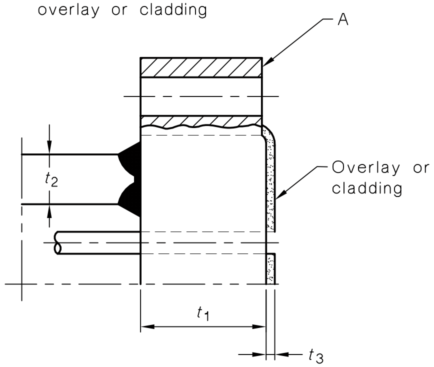
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(iv) Forged tube plate with stub 	AW	Max. of t_2, and $t_f/4$ in Figure 2.6.2(B)	t_2	t_2	—
	PWHT	Max. of ($t_1/4$) and t_2	t_2	t_2	—
(v) Tube plate with stubs 	AW	Max. of t_2; t_3; and $t_1/4$ in Figure 2.6.2(B)	t_2 (t_3)	t_2 (t_3)	—
	PWHT	Max. of $t_1/4$, t_2 and t_3	t_2 (t_3)	t_2 (t_3)	—

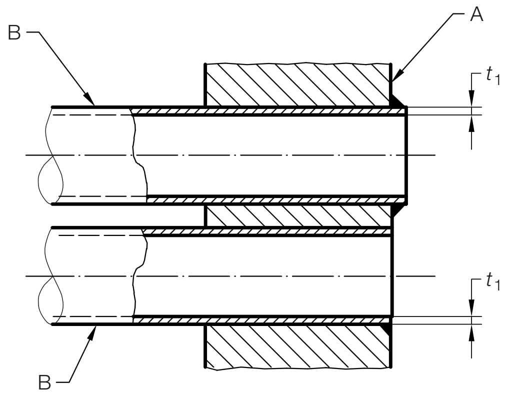
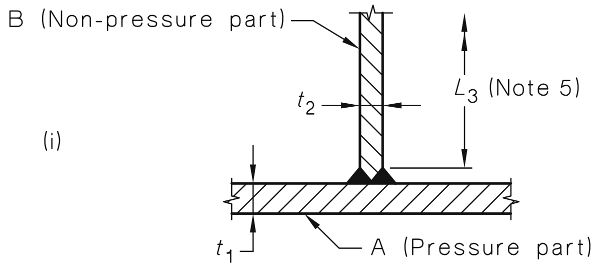
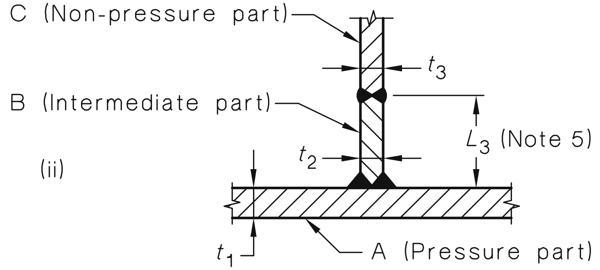
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(vi) Tube with long stub 	AW, where L<4t_2	Max. of t_1/4 and t_2	t_2	t_2	—
	AW, where L ≥4t_2	Max. of t_2 and (t_1/4) in Figure 2.6.2(B)	t_2	t_2	—
	PWHT	Max. of (t_1/4) and t_2	t_2	t_2	—
(vii) Tube plate with dissimilar overlay or cladding 	AW	Max. of (t_1+t_3)/4 and t_2	t_2 (Overlay t_3)	t_2	—
	PWHT	Max. of (t_1+t_3)/4 and t_2	t_2 (Overlay t_3 in Figure 2.6.2(A))	t_2	—

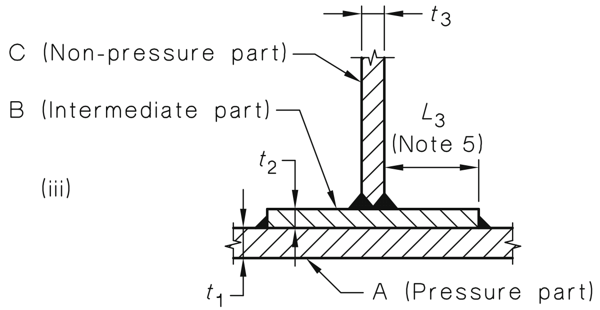
(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	**Weld**	**Part B**	**Part C**
(viii) Tube to tube plate connection 	AW	t_R not affected by tube weld	t_1	t_1	—
	PWHT	t_R not affected by tube weld	t_1	t_1	—
(g) Attachments 	AW	t_1	t_2	Lesser of t_1 and t_2	—
	PWHT	t_1	t_2	Lesser of t_1 and t_2	—
	AW	t_1	t_2	Lesser of t_1 and t_2	$t_3/4$
	PWHT	t_1	t_2	Lesser of t_1 and t_2	$t_3/4$

(*continued*)

TABLE 2.6.4 (*continued*)

Construction details	Condition (Note 1)	Reference thickness, t_R (Note 2)			
		Part A	Weld	Part B	Part C
(iii) 	AW	t_1	—	Lesser of t_1 and t_2	$t_3/4$
	PWHT	t_1	—	Lesser of t_1 and t_2	$t_3/4$

LEGEND:
AW = As-welded condition
PWHT = Post-weld heat treated condition

NOTES TO TABLE 2.6.4:

1 Unless noted, use Figure 2.6.2(A) for as-welded condition and use Figure 2.6.2(B) for post-welded heat treated condition.

2 t_R is only for the part or joint and condition shown assuming parts are seamless. Other welds also need to be considered.

3 For tapered flange take t_f = mean thickness.

4 For 1H, maximum slope is 1:4. For Class 1, maximum slope is 1:3.

5 L_3 = minimum length with suitable low temperature properties.
= greater of 50 mm and $2t_3$.

2.6.5 Impact testing

2.6.5.1 *When required*

Parent metal of pressure parts and of non-pressure parts directly attached by welding to pressure parts shall be impact-tested as required by Table 2.6.3.

Impact testing is not required for material of Group A to E steels thinner than 3 mm or where it is impracticable to obtain 10 mm × 2.5 mm Charpy V-notch specimens (see also Clause 2.6.2.3). See Clause 2.6.2.4 for other impact test exemptions.

For Group A to E steels impact testing is not required for 10 mm and thinner material provided the material design minimum temperature as calculated by Clause 2.6.3.2 is not colder than the temperature shown in Table 2.6.5.1.

TABLE 2.6.5.1

MATERIAL DESIGN MINIMUM TEMPERATURE

Thickness mm	As welded °C	PWHT °C
10	−15	−30
8	−20	−35
6	−25	−40
4	−40	−55
≤ 2	−55	−70

Certified reports of impact tests made by the material manufacturer shall be accepted as evidence the material complies with the requirements of this Clause, provided—

(a) the test specimens are representative of the material supplied and the material is not subject to heat treatment during or following fabrication, which will materially reduce its impact properties; or

(b) the materials from which the test specimens are removed were heat-treated separately such that they are representative of the material in the finished vessel.

The manufacturer of the vessel may have impact tests made to prove the suitability of a material that the materials manufacturer has not impact-tested, provided the number of tests and the selection of the test specimens is as specified in the material Standard.

2.6.5.2 *Test method*

Impact testing shall be in accordance with AS 1544.2 except—

(a) lateral expansion tests shall be in accordance with ASTM A 370 (see Table 2.6.3 for use); and

(b) the dropweight test for determining the NDTT shall comply with AS 2205.7.2 (see Table 2.6.3 for use).

2.6.5.3 *Test specimens*

Test specimens shall be selected and prepared as follows:

(a) *Number of Charpy V specimens* The number and location of Charpy impact test specimens shall be selected to adequately represent the material used in the vessel, and such selection shall be in accordance with a specification appropriate to the product form, e.g.—

(i) plates .... AS 1548;

(ii) pipes and tubes .... ASTM A 524;

(iii) forgings .... ASTM A 350;

(iv) castings .... ASTM A 352;

(v) bolting..ASTM A 320; and

(vi) pipe fittings .... ASTM A 420.

From Group F and Group G steels at least three Charpy V specimens (refer to Clause 2.6.5.6(d) for retests and requirements for additional test specimens) shall be made from each plate as heat-treated, or from each heat of bars, pipe, tube, rolled sections, forged parts or castings included in any one heat treatment lot. The specimens for plate shall be oriented transverse to the final direction of rolling. For circular forgings, the specimens shall be oriented tangential to the circumference, and for pipes and tubes the specimens shall be oriented longitudinally.

For Group A to E steels, plate specimens may be oriented longitudinally or transversely to the final direction of rolling.

For wrought material, at least three Charpy specimens shall be cut with the specimen parallel to the principal direction of hot working.

The manufacturer of small components other than bolting, either cast or forged, may certify a lot of more than 20 duplicate parts by reporting the results of one set of impact specimens taken from one such component selected at random, provided that the same specification and heat of material and the same process of production, including heat treatment, were used for all of the lot.

(b) *Dimensions of Charpy V specimens* Standard 10 mm × 10 mm specimens shall be used where the thickness or diameter permits. For material of nominal thickness 20 mm and over, 10 mm × 10 mm specimens shall not include material nearer to the surface than 3 mm. For material of nominal thickness under 20 mm, 10 mm × 10 mm specimens shall be machined so that they do not include material nearer to the surface than 1 mm. If the material is too thin to permit the preparation of 10 mm × 10 mm specimens, the dimension along the base of the notch shall be reduced to the largest possible of 7.5 mm, 5 mm and 2.5 mm.

The base of the notch shall be perpendicular to the original external surface.

(c) *Dropweight test specimens* Dropweight test specimens shall be selected as follows:

(i) For plate thicknesses 16 mm and over, one dropweight test (two specimens) shall be made for each plate as heat-treated.

(ii) For forgings and castings of 16 mm thickness or over, one dropweight test (two specimens) shall be made for each heat in any one treatment lot using the procedure in ASTM A 350 for forgings and in ASTM A 352 for castings.

2.6.5.4 *Impact test requirements*

Where impact tests are required by Clause 2.6.5.1, the test results shall comply with the criteria (test method and values) specified in Table 2.6.3 and the following:

(a) *General* General requirements for impact tests are as follows:

(i) Where Charpy V impact values are specified in Table 2.6.3, the average impact energy values of the three 10 mm × 10 mm Charpy V specimens shall be no less than the values given in Table 2.6.3 to satisfy the T_R and the values for individual specimens shall be no less than 70% of the specified minimum average value.

(ii) Where lateral expansion values are specified in Table 2.6.3, each specimen shall show 0.38 mm minimum lateral expansion on the opposite side of the notch for material up to 30 mm thick, and 0.60 mm for material over 75 mm thick, with linear interpolation.

(iii) When the nil ductility transition temperature (NDTT) is specified in Table 2.6.3, the NDTT shall be equal to or less than required T_R.

NOTE: The impact energy at a particular temperature is usually appreciably lower for test pieces cut transverse to the grain (i.e. transverse to the direction of principal hot working) than for pieces cut in the direction of the grain. Where test pieces must be cut transverse to the grain, the specified minimum impact energy for longitudinal specimens should be reduced. Where appropriate values are not specified in material specifications, requirements for transverse specimens should be a matter for agreement between the parties concerned.

(b) *Weld neck flanges, plate and slip-on (or hubbed) flanges, tubeplates and flat ends* The minimum impact energy shall comply with the requirements of Clause 2.6.5.4(a) using the appropriate T_R except that in no case shall the impact test requirements be less than those which would be required if they were unwelded.

The minimum impact energy for tubeplates having tubes attached by welding shall be derived in accordance with Clause 2.6.4(b), except that in no case shall the impact requirements for the tubeplates be less than those required for the tubes.

(c) *Attachments* The minimum impact energy for a non-pressure attachment welded directly to a pressure component shall be not less than that required for the pressure component to which it is attached.

2.6.5.5 *Impact test requirements for sub-sized specimens*

If the base material is less than 10 mm thick, the energy for sub-sized Charpy V specimens (i.e. less than 10 mm × 10 mm) shall be not less than values given in Table 2.6.2 and Table 2.6.3 times the appropriate equivalent energy factor given in Table 2.6.5.5(A).

TABLE 2.6.5.5(A)

EQUIVALENT ENERGY FACTORS FOR SUBSIDIARY TEST SPECIMENS

Thickness of test specimen mm	Equivalent energy factor
10 (standard)	1.0
7.5	0.8
5.0	0.7
2.5	0.35

NOTE: For test specimens between the above thicknesses, linear interpolation permitted.

Where a component shape does not permit a Charpy specimen size representative of the component thickness, the sub-sized specimen shall be tested at a temperature lower than that for a fully representative specimen by the temperature shift shown in Table 2.6.5.5(B).

TABLE 2.6.5.5(B)

TEST TEMPERATURE FOR SUB-SIZED SPECIMENS EXTRACTED FROM THICKER SECTIONS

Specimen size representative of component thickness tested at T_{CV} (See Note 1) mm	Sub-sized specimen	
	Size mm	Test temperature °C
10 × 10	7.5 × 10	T_{CV} − 5
	5.0 × 10	T_{CV} − 20
	2.5 × 10	T_{CV} − 30
7.5 × 10	5.0 × 10	T_{CV} − 15
	2.5 × 10	T_{CV} − 15
5.0 × 10	2.5 × 10	T_{CV} − 10

NOTE: T_{CV} = Standard test temperature in Table 2.6.2.

2.6.5.6 *Retests*

According to the nature of failure of a test, retests may be performed. The following provisions apply:

(a) *Failure of one specimen* If the average of the three Charpy impact tests exceeds the specified minimum average energy value specified in Table 2.6.3 but one test piece fails to give the specified minimum individual value, three additional test pieces from the original sample shall be tested. The result shall be added to those previously obtained and a new average calculated. If the average value of the six tests is not less than the specified minimum average, and not more than one result of the six tests is below the specified individual test value, then the product complies with this Clause (2.6.5).

(b) *Failure of average of tests* If the average of the three impact tests fails to attain the specified minimum average energy value or if two of the tests fall below the specified minimum individual value, the material represented shall be deemed not to comply with this Clause (2.6.5).

(c) *Failure due to specimen defect or procedure error* Where failure is the result of a defect peculiar to the specimen or to an error in the test procedure, the result shall be discarded and a further specimen substituted.

(d) *Failure in lateral expansion test for all size specimens* If the value of the lateral expansion for one specimen is below 0.38 mm but not below 0.25 mm, and the average value for the three specimens equals or exceeds 0.38 mm, then a retest of three additional specimens may be made, each of which shall attain values equal to or exceeding 0.38 mm. If the required values are not obtained in the retest or if the values in the initial test are below the minimum required for retest, the material shall be either rejected or submitted to a further heat treatment. After such re-heat treatment, three specimens shall be tested and the lateral expansion for each shall equal or exceed 0.38 mm.

(e) *Failure in the dropweight test* If one of the two test specimens fails to meet the no-break criterion then two more specimens shall be taken and retested. Each of these specimens shall meet the no-break criterion. If this criterion is not met in the retest the material shall be rejected or submitted to a further heat treatment. After such re-heat treatment, two specimens shall be tested and the no-break criterion shall be met.

2.6.6 Fracture mechanics analysis

2.6.6.1 *General*

A fracture mechanics analysis may be used in determining the suitability of materials and vessels for their intended application when one or more of the following apply:

(a) The materials are not covered by the material or application Standard.

(b) The requirements of Clause 2.6.1, Method 1 cannot be adhered to.

(c) The imperfections are outside the non-destructive testing requirements.

2.6.6.2 *Analysis*

The analysis shall be carried out in accordance with methods covered by API 579, BS 7910 or equivalent, or by AS/NZS 3788 where thickness ≤ 30 mm. The analysis shall be fully documented.

2.6.6.3 *Material tests*

Fracture toughness properties shall be determined by suitable methods. The properties should be obtained with fracture testing procedures using full thickness single edge notched bend specimens or equivalent compact tension tests with fatigue cracks located through thickness in the weld-centre line and in the parent metal. Further test sampling of the heat affected zone may also be specified, particularly when fatigue or other in-service crack growth mechanisms may be significant.

When heat-affected zone tests are specified, special considerations are necessary with regard to the placement of the notch and metallurgical sectioning subsequent to testing.

2.6.6.4 *Required fracture toughness*

The fracture toughness required shall be based on the following:

(a) An imperfection size equal to one of the following:

  (i) A through-wall total length of 10 mm, or a quarter wall thickness surface imperfection with length six times its depth.

  (ii) One weld bead depth with length six times its depth.

  (iii) Dimensions selected by the manufacturer, and agreed by the parties concerned.

(b) An equivalent stress (or strain) associated with the hydrostatic test condition, for an imperfection in a region of stress concentration and subject to residual stresses equivalent to the ambient temperature yield strength of the material for as-welded components, or 30% of the yield for components that are unwelded or postweld heat treated.

2.6.6.5 *Non-destructive examination methods*

Non-destructive examination shall be used to accurately determine the size of imperfections to ensure they fall within the limits of Clause 2.6.6.4.

NOTES:

1 See AS 4037 for more sensitive methods and for time of flight diffraction ultrasonic examination.

2 When radiography is used, a full thickness imperfection should be assumed.

2.6.7 Non-metallic materials

Non-metallic gaskets, packing, supports and similar parts used for low temperature service shall be suitable for the service at the T_{min} and allowance shall be made for any hardening or embrittlement.

2.7 MATERIAL FOR HIGH TEMPERATURE SERVICE

2.7.1 General

Materials of pressure parts of vessels shall not be used at operating temperatures in excess of the highest design temperature for which strengths are shown in Table B1 for the given material, except where the material is shown to be suitable for the service conditions and is acceptable to the parties concerned.

2.7.2 Selection of materials for high temperature service

In selecting materials for prolonged exposure to high temperatures, consideration shall be given to each of the following factors:

(a) The loss of thickness due to scaling.

(b) The possible graphitization of carbon, carbon-manganese and carbon-silicon steels at temperatures above 425°C and of carbon-molybdenum steel at temperatures above 470°C.

(c) The embrittlement of high alloy steel Type 430 at temperatures above 425°C.

(d) Other exposure effects on materials (e.g. reduction in metal toughness due to long term hydrogen damage; age embrittlement of steels; sigma phase embrittlement of stainless steels; or long term HAZ cracking in some Cr-alloy steels over 500°C).

NOTE: Some piping standards have introduced requirements for reduction of design strength for welded components, of certain materials, operating in the creep range. Guidance on a conservative approach can be found in AS 4041 or ASME B31.3.

(e) The reliability of elevated temperature test data and the applicability of the design stress basis given in Appendix A.

(f) Reduction in material strength and rupture pressure due to accidental fire conditions.

2.7.3 Valves and similar components

The maximum operating temperatures of valves and similar components may be limited by the trim material.

2.7.4 Brazing and soldering materials

The operating temperature shall not exceed 95°C for brazing materials and 50°C for soldering materials except higher temperatures may be used when agreed between the parties concerned and qualified by suitable tests (see AS/NZS 3992).

2.7.5 Steels

Steels for vessels for use at temperatures above 50°C may be supplied and used with or without elevated temperature properties being verified or hot-tested by the material manufacturer.

NOTE: Testing should be in accordance with AS 2291, or another recognized method.

The use of cladding or lining with unstabilized chromium-alloy stainless steel with chromium content over 14% is not recommended for design temperatures over 425°C.

2.8 NON-DESTRUCTIVE TESTING OF MATERIALS

Where increased assurance of material quality is required to assist in economic manufacture, e.g. in tubeplates or major components of Classes 1H, 2H, 1S, 2S and 1 vessels, non-destructive testing should be carried out on material before fabrication, as required by the manufacturer or by the purchaser (see Appendix E).

Where ultrasonic examination of welded joints is required (see AS 4037), consideration shall be given to the need for ultrasonic examination of the parent material in the vicinity of the area to be welded, to ensure this portion of the parent material is sufficiently free from defects that would prevent adequate ultrasonic inspection of the joint. This may be done by the use of parent material that has been ultrasonically examined by the material manufacturer or locally by the vessel manufacturer prior to welding. Similarly, attention should be given to plate for applications where there is a high stress gradient through the plate thickness, e.g. at set-on nozzles.

Where high casting quality factors are required, castings shall meet the NDE requirements of AS 4037.

All forgings 100 mm and over in thickness shall be ultrasonically examined in accordance with, and shall meet the requirements of, Section 16 of AS 4037.

2.9 MATERIALS FOR CORROSIVE SERVICE

In selecting materials for exposure to corrosive conditions, the following shall be considered:

(a) Satisfactory past performance of the material under similar design and service conditions.

(b) Selection of a high corrosion resistant material, provision for corrosion allowance (see Clauses 3.2.4.2 and 3.5.3.3) or use of a corrosion resistant lining or cladding (see Clauses 3.2.4.4).

Performance tests are advisable if, because of lack of experience, the performance is not known. It is advisable the frequency of in-service inspection is increased or monitoring is performed until there is sufficient experience to justify that satisfactory performance will be achieved.

When hot dip galvanizing is applied, the material shall be subject to appropriate qualification testing. Materials shall not be heat treated after galvanizing. Examples include high strength steels, steels with a high silicon content or heat affected zones.

NOTE: Corrosion prevention practice should also be considered. Guidance is given in Appendix D.

SECTION 3 DESIGN

3.1 GENERAL

3.1.1 Main design requirements

The design of vessels and vessel parts subjected to pressure shall comply with the requirements of this Section.

This Standard only deals with the mechanical design of vessels and assumes that a process design and piping and instrument diagram for the vessel are available to the designer.

Where the process design and piping and instrument diagrams are not available, any assumptions made by the designer shall be stated and are the subject of agreement between the parties concerned.

There shall be a design specification covering vessel and service requirements and other essential requirements. This specification is the subject of agreement between the purchaser and designer/manufacturer.

3.1.2 Design responsibility

The design of the vessel shall be in accordance with this Section and in accordance with the design conditions specified by the purchaser. (See Appendix E for guidance on information to be supplied). Where the design conditions are not fully specified, the designer shall make suitable assumptions to determine the design conditions. Any such assumptions shall be suitably documented.

The designer shall be suitably qualified and experienced, and be competent in the design of pressure vessels. (See Appendix P for guidance on competency of personnel).

Hazards associated with operation of the vessel shall be identified, including the likelihood and possible consequences of the failure of the vessel, and the identified hazards shall be assessed and suitably controlled. This hazard assessment shall include (but is not necessarily be restricted to) consideration of the following elements:

(a) The adequacy of materials, design, manufacture, operation, inspection and maintenance.

(b) The nature of service conditions.

(c) The pressure energy (pressure and volume) of the vessel.

(d) The nature of contents when released.

(e) The location with respect to people and the plant.

(f) Where appropriate, the economics of repair, replacement and obsolescence.

NOTE: For guidance on the management of risk, see Appendix C.

3.1.3 Design methods

Vessels shall be designed by one or more of the following:

(a) Design by formulae and equations and related requirements given in this Standard (see Section 3).

(b) Where a design method is not provided in this Standard, a design method in another internationally recognized standard may be used.

(c) Design by analysis using rigorous mathematical stress analysis such as linear elasticity theory or Finite Element Analysis (FEA) (see Appendix H or I).

(d) Design using experimental stress analysis, e.g. strain gauging, photoelasticity, etc. (see Clauses 5.12.1 to 5.12.6).

(e) Design by fracture mechanics, according to either BS 7910 or API 579.

(f) Design using destructive or proof type testing (see Clause 5.12.7).

(g) Design based on successful experience of equivalent designs under equivalent conditions.

When applying the design methods in Items (b) to (f), the minimum calculated thickness shall comply with the requirements of Appendices A, H and M of this Standard.

3.1.4 Design against failure

The designer shall consider all the failure modes listed in Appendix G, and design against the failure modes that are feasible for that vessel under all credible loads in manufacturing, transport, installation and service conditions, including normal operation, start up, shut down tests and process upset. In special conditions the failure mode may influence the risks encountered during failure.

NOTE: It is intended that compliance with this Standard will satisfy the above requirement.

The design strengths given in Table B1 of this Standard have been selected to satisfy requirements to design against excessive deformation, ductile fracture, and creep.

For design against the following failure modes, refer to the appropriate clauses in this Standard:

Instability (buckling) .... Clause 3.9.

Corrosion and wastage.............................. Clause 2.9, 3.2.4 and Appendix D.

Fatigue .... Appendix M.

Brittle Fracture .... Clause 2.6 and 3.2.5.

High temperature metallurgical damage ..... Clause 2.7.

3.1.5 Design against excessive deflection

Where necessary, design shall avoid excessive deflection or distortion that could result in the following:

(a) Leakage or loss of containment of QA doors.

(b) Loss of adequate support, e.g. trays.

(c) Inadequate draining, e.g. between supports on long vessels.

(d) Cracking of brittle coatings or components.

(e) Interference or inadequate or excessive clearance with adjacent parts.

NOTE: Such deflection may arise due to vibration.

3.1.6 Design criteria for Class 1H and 2H vessels

The minimum thickness of vessel parts subject only to fluid pressure shall be calculated from the equations given in Clauses 3.7 to 3.13.

Where parts of the vessel are subject to loads in addition to that of fluid pressure, an equivalent stress intensity based on the maximum shear stress yield criterion shall be calculated. For loads giving rise to primary membrane stresses, the equivalent stress intensity shall not exceed the design strength at the design temperature (see Clause 3.3) except as allowed for in Table 3.1.6.

Where in addition to the primary membrane stress there are other stresses present, the equivalent stress intensity at any location shall not exceed the stress intensity limits given in Appendix H (see Figure H1). Clauses 3.7 to 3.13 might not ensure against fatigue failure. Thus reference shall be made to Appendix M to determine when the above Clauses are applicable or when recourse to further fatigue or other analysis is required.

Irrespective of whether credit is taken or is not taken for cladding in the computations for the dimensions of components for integrally clad vessels designed for operation at other than ambient temperature, calculation of the primary membrane stresses in both the base material and the cladding shall be performed and shall take into account any differential coefficients of thermal expansion, and through-thickness temperature gradients during starting and shutdown operations. The calculated stresses shall not exceed the relevant design strengths given in Table B1.

TABLE 3.1.6

MEMBRANE STRESS INTENSITY LIMITS FOR VARIOUS LOAD COMBINATIONS

Condition	Load combination		Membrane stress intensity limit (kf) (see Note 1 and 2)	Calculated stress limit basis
Design	A	The design pressure, the dead load of the vessel, the contents of the vessel, the imposed load of any mechanical equipment, and external attachment loads	$1.0f$	Based on the corroded thickness at design metal temperature
	B	Condition A above plus wind forces	$1.2f$	Based on the corroded thickness at design metal temperature
	C	Condition A above plus earthquake forces NOTE: The condition of structural instability or buckling must be considered	$1.2f$	Based on the corroded thickness at design metal temperature
Operation	A	The actual operating loading conditions. This is the basis of fatigue life evaluation	See Appendix M	Based on corroded thickness at operating pressure and operating metal temperature
Test	A	The required test pressure, the dead load of the vessel, the contents of the vessel, the imposed load of any mechanical equipment, and external attachment loads	See Clause 5.10. for hydrostatic test, and Clause 5.11 for pneumatic test	See Clause 5.10 for hydrostatic test, and Clause 5.11 for pneumatic test

NOTES:

1 f is the design strength at the design temperature as determined by Clause 3.3.1.

2 k is a load factor.

3.2 DESIGN CONDITIONS

3.2.1 Design and calculation pressures

3.2.1.1 *Design pressure of vessel*

The design pressure (see Clause 1.8) shall be the pressure specified by the purchaser, the application specification or as otherwise determined in accordance with this Standard. See also Figure 3.2.1 and Clause 3.2.1.4.

The design pressure shall not be less than the set pressure of the lowest set pressure-relief device.

In selecting the design pressure, a suitable margin, above the maximum operating pressure (see Clause 1.8) should be made to allow for probable surges of pressure during operation and to prevent unnecessary operation of pressure-relief devices. Where pressure relief devices are used, the design pressure is often assumed to be 5% to 10% above the operating pressure at the most severe condition, but where wide surges in pressure and temperature may occur, this margin may need to be increased. Where bursting discs are used, it is recommended the design pressure of the vessel be sufficiently above the normal operating pressure to provide a sufficient margin between operating pressure and bursting pressure to prevent premature failure of the bursting disc (refer to AS 1358).

3.2.1.2 *Calculation pressure of a vessel part*

A vessel part shall be designed for the most severe conditions of coincident pressure and metal temperature expected in normal operation, excluding the excess pressures developed during the hydrostatic test or during operation of pressure-relief devices. The vessel design shall also be suitable for the test fluid and vessel position during the hydrostatic pressure test. The most severe condition of coincident pressure and temperature shall be that condition which results in the greatest thickness of the part of the vessel under consideration, not including corrosion allowance. The pressure and temperature at this condition, with a suitable margin (see Clause 3.2.1.1), shall be used as the calculation pressure and temperature. For a large or complex vessel, there may be different calculation pressures and temperatures for various parts of the vessel.

In determining the calculation pressure of a part, provision shall be made for pressures due to static head of contained liquids or pressure differentials due to fluid flow. The calculation pressure of any part using the actual thickness minus any corrosion allowance and adjusting for any difference in static head, or pressure differential, or in temperature, or any combination of these that may occur under the least favourable conditions shall at least equal the design pressure of the vessel.

3.2.1.3 *External pressure*

NOTE: Pressure is considered to be external when it acts on the convex surface of a cylindrical or spherical part of the vessel, tending to cause collapse.

For vessels or vessel parts subjected to vacuum conditions or external pressure or different pressures on opposite sides of the part under consideration, the calculation pressure shall be the maximum differential pressure that the vessel part may be subjected to at the most severe condition of temperature and differential pressure considering possible loss of pressure on either side of the part in question. Where relevant, the calculation pressure shall provide for the self-weight of the vessel part which shall be based on actual plate thickness, including corrosion allowance.

For vessels subject internally to vacuum only, the external design pressure shall be 101 kPa or be 25% more than the maximum possible external pressure, whichever is smaller. Where the design pressure is less than 101 kPa, the vessel shall be provided with a vacuum relief device or hydraulic seal of an appropriate reliable type. (See Clause 8.10 for setting of vacuum relief devices.)

Where each of the following conditions apply to a vacuum vessel, the calculation pressure may be reduced to two-thirds of the external design pressure (as a means of reducing the nominal factor of safety for the shell, ends and stiffening rings from three to two):

(a) Buckling of the vessel will not cause a safety hazard.

(b) The vessel forms a vacuum jacket on another vessel and buckling of the jacket will not lead to failure of the inner vessel or support structure.

(c) The vessel does not support walkways or operational platforms higher than 2 m above grade.

(d) The vessel is single wall type and the contents are not lethal (according to AS 4343) and is not greater than 5 m in height.

(e) Loading points and lifting lugs are designed and located so as to avoid buckling.

(f) Checks on the circularity and shape of the vessel are carried out and confirm compliance with AS 4458.

Allowance shall also be made for the vacuum conditions that may arise with certain vessels that are normally under internal pressure, e.g. vessels containing steam or vapour which condenses at low ambient temperatures.

3.2.1.4 *Design pressure for liquefied gas vessels*

For vessels subject to pressurization by liquefiable gases, the design pressure shall, in the absence of design requirements in the relevant application Standard, be the greater of the following:

(a) The pressure produced by the most severe operating conditions, excluding fire or other abnormal circumstances.

(b) The vapour pressure of the liquid content at the maximum service temperature in accordance with Clause 3.2.2.4. Allowance shall be made for the partial pressures of other gases or impurities in the vessel that might increase the total pressure.

The design shall also ensure that at the maximum service temperature, the filling ratio is such that the liquid phase on thermal expansion does not completely fill the vessel and the vapour space is not compressed to such an extent that partial pressure of inert gases causes venting through safety valves.

The filling ratio is the ratio of the mass of gas in a vessel to the mass of water the vessel will hold. The maximum filling ratio should be obtained (where given) from the relevant application code for a given gas, e.g. AS/NZS 1596.

3.2.2 Design and service temperatures

3.2.2.1 *Design temperature for other than clad vessels*

The design temperature of other than clad vessels shall be taken as the metal temperature which, with the coincident calculation pressure, results in the greatest thickness of the part under consideration. It shall be taken as not less than the metal temperature reached at the mean wall thickness when the part is at the calculation pressure.

The metal temperature at the mean wall thickness shall be taken as the temperature of the contained or surrounding fluid, as appropriate and shall comply with Table 3.2.2.1, except where calculations, tests, or previous service and experience support the use of another temperature.

For design against brittle fracture, the minimum operating temperature shall be used as a basis. (See Clause 2.6.3.)

Appropriate allowance shall be made for feasible loss of refractory or insulation.

TABLE 3.2.2.1

DESIGN TEMPERATURE FOR HEATED PARTS

	Type of heating	Design temperature for heated parts (unless measured or calculated) (see Notes 1 and 2)
1	By gas, steam or liquid	The highest temperature of the heating medium (Note 3)
2	Directly by fire, exhaust gas, or electric power	For protected parts or parts heated primarily by convective heat, the highest temperature of the parts contents plus 20°C For parts not protected from radiation, the highest temperature of the parts contents plus the greater of 50°C and 4 × part thickness (mm) + 15°C, and with a minimum temperature for water of 150°C
3	By indirect electric power i.e. electrode or element in water (Note 4)	The highest temperature of the vessel contents
4	By solar radiation without protection of parts	(a) Direct: 50°C for metals; measure for non-metals (b) Focussed: As measured or calculated

NOTES:

1 Measurement where practicable with embedded and protected thermocouples is recommended. See AS 1228 for typical calculation of parts exposed to fire.

2 Provision shall be made for limited heat absorption rates with some fluids and for feasible deviation from ideal temperatures e.g. due to restricted flow in some tubes, loss of baffles or shields, abnormal firing conditions with new fuels and equipment, fouling, excessive firing rates, rapid starts, or poor circulation or mixing.

3 For tubular and plate heat exchangers and similar vessels, a lower temperature determined by heat transfer analysis may be used for the various parts provided suitable provision is made to cater for overheating in the event of loss or restricted flow of cold fluid. See AS 3857 for tube plate design.

4 Assumes pressure retaining parts are completely submerged in liquid and there is no radiation. See Clause 3.32 for special controls to prevent excessive wall temperature due to radiant heating in the event of element exposure with low fluid level.

3.2.2.2 *Design temperature for clad or lined vessels*

The design temperature for clad or lined vessels, where design calculations are based on the thickness of the base material exclusive of lining or cladding thickness, shall be taken as that applicable to the base material.

Where design calculations are based on the full thickness of clad plate (see Clause 3.3.1.2), the maximum design temperature shall be the lower of the values allowed for the base material or cladding material referenced in Table B1.

3.2.2.3 *Temperature fluctuations from normal conditions*

Where temperature fluctuations from normal conditions occur, the design temperatures in Clauses 3.2.2.1 and 3.2.2.2 need not be adjusted provided—

(a) the temperature is in the creep range (i.e. at a temperature where the stress to cause rupture or 1% strain in 100 000 h is the stress that determines the design strength);

(b) the average temperature during any one year of operation will not exceed the design temperature;

(c) normal fluctuations in temperature will not cause the operating temperature to exceed the design temperature by more than 15°C; and

(d) for steel components, abnormal fluctuations in temperature will not cause the operating temperature to exceed the design temperature by more than 20°C for a maximum of 400 h per year, or 35°C for a maximum of 80 h per year.

Where the maximum temperature will exceed these limits, the design temperature shall be increased by the amount of this excess.

Where the excess temperatures are likely to exceed the temperatures in Item (d) for more than 50% of the times shown, a temperature recorder shall be fitted.

NOTE: The purchaser is responsible for ensuring the recorder is fitted and operated to ensure the above requirements are fulfilled.

3.2.2.4 *Maximum service temperature for liquefied gas vessels*

The maximum service temperature shall be taken as the greater of the following:

(a) The maximum temperature to which the contents will be subjected by the process under the most severe operating conditions.

(b) The maximum temperature which the liquid contents are likely to attain due to ambient conditions.

NOTE: AS 2872 sets out a method for estimating the temperatures and corresponding pressures of fluids in vessels subject to atmospheric and solar heating in the hottest month of the year in various locations in Australia.

3.2.3 Design loadings

The loadings to be considered in the design of the vessel shall include the following, where relevant:

(a) Internal or external (or both) design pressures.

(b) Maximum static head of contained fluid under normal operating conditions and under any specified abnormal fluid levels above normal operating conditions, including the effect of fluids with a specific gravity greater than 1.

(c) The force due to standard gravity acting on the mass of the vessel and normal contents under operating and test conditions, including conditions of reduced or zero pressure, if applicable.

(d) Superimposed loads, such as other vessels, attached piping weight and operating loads, lining, insulation, operating equipment, platforms, snow, water, ice, and the like.

(e) Wind loads—See Appendix J for wind loads.

NOTES:

1 In calculating the adequacy of design for the hydrostatic test, only 75% of the normally applied wind load need be considered to act simultaneously with the other loadings.

2 For information on dynamic wind loads, refer to BS 4076, Moody, Mahajan, De Ghetto and Long, Freese and Bednar*.

(f) Earthquake loads—See Appendix J and AS/NZS 1200 for selection of earthquake loads.

NOTE: Wind and earthquake loads need not be assumed to occur simultaneously.

(g) For transportable vessels, the inertia forces and loads from the chassis or support frames due to motion during transport (see Clause 3.26).

* See Appendix R for bibliographic details.

(h) Local stresses at lugs, saddles, girders, supports and nozzles due to the reaction of vessel supports and loads from internal and external structures and connected piping, considering all creditable imposed loading acting concurrently.

NOTE: While the design of support structures needs to account for maximum concurrent loadings, the summation of specified or calculated maximum piping loads might be an unrealistic total loading condition and so lead to over-designed supports. The combination of piping loads (and other external applied loads) which represents the maximum realistic loading condition should be the subject of agreement between the parties concerned.

(i) Forces caused by the method of support during lifting, transit and erection.

(j) Shock loads due to changes in fluid flow, surging of contents, sloshing of fluids, or reaction forces (e.g. relief valve discharge).

(k) Moments due to eccentricity of the centre of pressure relative to the neutral axis of the section.

(l) Forces due to temperature conditions, including the effects of differential expansion of parts or attached piping.

(m) Other external or environmental conditions (e.g. flooding, wave action, impact, collision, or earth loads).

(n) Forces due to fluctuating pressure or temperature.

Formal analysis of the effect of loading for Items (i) to (n) is required only where it is not possible to demonstrate the adequacy of the design, e.g. by comparison with the behaviour of comparable vessels.

The conditions under which a fatigue analysis that takes into account loadings for Items (h) to (n) is not required are set out in Appendix M.

Where the vessel is required to be hydrostatically tested in the final installed position as part of periodic inspection or repair, the vessel, supports and foundations shall be designed for full hydrostatic loading unless alternative measures are taken. The design specification should state whether this is required or not. Where the vessel cannot be hydrostatically tested in-situ or special arrangements are required, the name plate or documentation may state this.

3.2.4 Corrosion, (including all forms of wastage)

3.2.4.1 *General*

Each vessel or part thereof liable to corrosion (see Clause 1.8) shall have provision made against corrosion for the desired life of the vessel to safeguard against the need for reduction in operating pressure, excessive repairs or replacement. See Appendix G for forms of corrosion.

This provision may consist of—

(a) a suitable increase in thickness of the material over that determined by the design equations to cover general corrosion (this may not be applicable where localized corrosion occurs) (see Clause 3.2.4.2);

(b) linings or wear or impingement plates;

(c) cathodic protection;

(d) chemical treatment of contained fluid;

(e) postweld heat treatment to avoid stress corrosion;

(f) selection of material suitably resistant to corrosion under the intended service conditions; or

(g) a combination of these or other suitable methods.

Where corrosion effects are known to be negligible or entirely absent, no provision need be made.

3.2.4.2 *Corrosion allowance*

Where provision for corrosion is to be made in accordance with Clause 3.2.4.1(a), the minimum calculated thickness shall be increased by an amount at least equal to the expected loss of wall thickness during the desired life of the vessel. See Appendix D for selection of corrosion allowance.

The dimensional symbols used in all design formulae throughout this Standard represent the dimensions in the corroded condition.

Corrosion may occur on both sides of the wall of some vessels and necessitate an allowance on both sides. Corrosion allowance need not be the same for all parts of the vessel where different rates of attack are expected.

In selecting the corrosion allowance, consideration shall be given to the type of wastage.

Carbon, carbon-manganese and low alloy steel vessels used for compressed air service, steam service or water service shall have a minimum of 0.75 mm corrosion allowance on each metal surface in contact with such fluid except that this allowance is not required where seamless cladding or lining, other suitable linings or specially dried air are used.

3.2.4.3 *Dissimilar metal corrosion*

Where dissimilar metals are used together in a corrosive environment, control of galvanic action by correct design procedure shall be instituted. This is particularly important with aluminium.

3.2.4.4 *Linings*

Vessels may be fully or partially lined with material resistant to corrosion. Such material may be loose, intermittently welded, integrally clad, sprayed or surface welded. Special provisions shall be made for vitreous enamel lining. (See BS 6374, Part 1 to Part 5 for recommended practice in lining vessels.)

Where such linings effectively prevent contact between the corrosive agent and the vessel base material during the life of the vessel, no corrosion allowance need be provided. Normally, such linings will include metal cladding, applied metal linings, glass lining and thick rubber or plastic linings. Paints, dip galvanizing, electro-deposits and sprayed metals are excluded unless specially agreed upon between the parties concerned.

Where corrosion of the cladding or lining material is expected, the cladding or lining thickness shall be increased by an amount that will provide the required service life of the vessel.

3.2.4.5 *Corrosion data*

It is not practicable to give more definite rules than those preceding to safeguard against the effects of corrosion because of its complex nature and the many combinations of corrosive environments and materials. However, additional data are given in Appendix D as a guide.

3.2.5 Low temperature service

The following recommendations apply to vessels made of ferritic steel and with Design Minimum Temperature colder than 0°C:

(a) Sufficient flexibility should be provided to cater for differential expansion or contraction.

(b) The vessel should be of simple configuration.

(c) The occurrence of rapid changes in temperature likely to give rise to severe temperature gradients should be avoided. Where this situation nevertheless occurs consideration should be given to special design details. A typical desirable design detail is given in Figure 3.2.5.

(d) Care should be taken to avoid details that will produce local areas of high stress, e.g. lugs, gussets producing discontinuous stiffening and abrupt structural changes.

(e) Discontinuous stiffeners or continuous stiffeners attached by intermittent welding should not be used.

(f) Doubling plates should be used in attaching vessel supports.

(g) Pipe supports and anchors should be attached to an encircling mechanically separate sleeve.

(h) Screwed connections and socket-welded valves and fittings should not be used.

(i) Nozzles and complicated structural attachments should be welded to shell plates in the workshop and be considered as a separate sub-assembly which may also be evaluated individually with regard to the desirability of a separate heat treatment.

(j) Non-pressure parts should be attached to pressure parts via intermediate parts.

Where an intermediate part is used between a pressure and non-pressure part, it shall be subject to the same restrictions as the pressure part to which it is attached. The requirements
A2 of this provision shall apply over a distance of at least $2t_2$ or 50 mm, whichever is the greater (see Table 2.6.4, Item (g) Attachments).

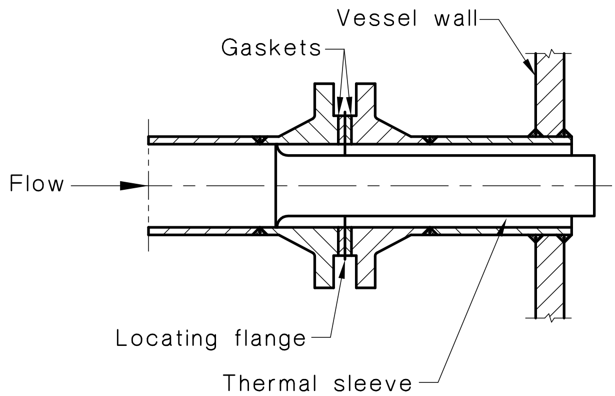


FIGURE 3.2.5 EXAMPLE OF THERMAL SLEEVE TO AVOID SEVERE THERMAL GRADIENTS

3.2.6 Vessel life

3.2.6.1 *General*

Vessels or components shall be designed for an appropriate life against deterioration by time-dependent modes of failure such as corrosion, fatigue, creep or combinations thereof.

For design against corrosion (including all forms of wastage), see Clause 3.2.4. For design against fatigue under severe cyclic stresses see Appendix M. For design against creep for a specific design lifetime, see Clause 3.2.6.2.

3.2.6.2 *Design lifetime in creep (high temperature) range*

The design strengths given in Table B1 for design temperatures in the creep range apply for a nominally indefinite component lifetime. However, each component whose design temperature is such that the applicable design strength is time-dependent may be designed for an appropriate, agreed lifetime, using the basis given in Paragraph A10 of Appendix A and data for different lifetimes in the relevant material specification or in AS 1228 or PD 5500. It is not intended that the same lifetime necessarily be adopted for all components. Replaceable components may be designed for shorter lifetimes than the general life expectancy of the vessel.

NOTE: The design lifetime of each component is a matter for agreement between the parties concerned.

No component designed on the basis of time-dependent material properties shall remain in service beyond the agreed design lifetime unless a review is made of its continued fitness for service based on inspection for creep damage and consideration of its temperature/stress history and the latest material data. Subject to satisfactory periodic review, service life may be extended beyond the agreed design lifetime.

In the above review, particular attention shall be paid to geometrical discontinuities and details subject to load or temperature cycling. Consideration shall be given to the installation of suitable equipment to record and provide a time-temperature history and a time-pressure history of the component. Additionally, it is recommended that the dimensional changes due to creep be recorded periodically to assist the review. Metallurgical examination and short-term creep rupture testing may be useful.

NOTE: Documents such as BS PD 6510 (obsolescent) and ANSI/API RP 530 provide examples of procedures followed.

3.2.7 Change in design conditions

A vessel or vessel component may be used at pressures or temperatures greater than the original design conditions and greater than permitted in Clause 3.2.2.3 provided that all of the following conditions are met:

(a) Each component so affected complies with the requirements of this Standard.

(b) The time at higher pressure or temperature does not reduce the proposed new design lifetime by more than 5%.

(c) Safety valves and other protective devices comply with the requirements of this Standard at the new design conditions.

(d) The parties concerned agree.

3.3 DESIGN STRENGTHS

3.3.1 Design tensile strength (*f*)

3.3.1.1 *General*

The design tensile strengths to be used with the equations in this Standard are given in Table B1 for materials other than bolting, and Table B2 for bolting.

For variations to the design tensile strengths given in Tables B1 and B2, see—

(a) Clause 3.21.1 for bolted flanged connections;

(b) Clause 3.26 for transportable vessels;

(c) Paragraph L3.3 for cryogenic vessels; and

(d) Clause 3.33 for vessels with increased tensile strength at low temperatures.

Design tensile strengths for materials not listed in Table B1 and B2 shall be determined in accordance with Appendix A.

NOTE: It is recognized that bending or local peak stresses in pressure vessels may exceed the strength values given in Table B1. When such stresses are to be calculated, criteria given in Appendix H may be followed when dealing with ductile materials (but using the values of f in Table B1)

To those design strengths the following shall be applied where appropriate:

(i) Welded joint efficiency (see Clause 3.5.1.7).

(ii) Brazed joint efficiency (see Clause 3.5.3).

(iii) Ligament efficiency (see Clause 3.6).

(iv) Casting quality factor shall be taken as one of the following:

(A) Carbon, carbon-manganese, low alloy and high alloy steel castings ....0.80.

(B) Non-ferrous and nodular iron castings ....0.80.

(C) For Items (A) and (B), where justified by additional testing to AS 4037 .....0.90.

(D) Iron castings covered in Clause 2.5.3.1 ....0.80.

For some vessels operating under special conditions and as required by the designer, it may be desirable to adopt reduced design strength to—

(1) limit deflection in close fitting assemblies;

(2) allow for abnormal fatigue, corrosion fatigue or stress corrosion conditions (see Clause 3.1.4);

(3) allow for exceptionally long life; or

(4) provide for other design conditions not intended to be covered by the design strength criteria in Appendix A.

3.3.1.2 *Design tensile strength for clad and lined material*

The following requirements apply:

(a) *Applied corrosion-resistant linings* The thickness of material used for applied lining shall not be included in the computation for the required wall thickness of a lined vessel. The design strength shall be that given for the base material in Table B1, at the design temperature (see Clause 3.2.2.2).

(b) *Weld overlay or integrally clad plate without credit for full cladding thickness* Except as permitted in Item (c), design calculations shall be based on the total thickness of the clad plates less the specified nominal minimum thickness of cladding. A reasonable excess thickness either of the actual cladding or of the same thickness of corrosion-resistant weld metal may be included in the design calculations as an equal thickness of base material. The design strength value shall be that given for the base material in Table B1, at the design temperature (see Clause 3.2.2.2).

(c) *Weld overlay or integrally clad plate with credit for cladding thickness* Where clad plate complies with ASTM A 263, A 264 or A 265 and the joints are completed by deposition of corrosion-resisting weld metal over the weld in the base material to restore the cladding, the design calculation may be based on the use of the design strength for the base material listed in Table B1, using a total thickness equal to—

$$t = t_{\mathrm{base}} + t_{\mathrm{clad}} \times \frac{f_{\mathrm{clad}}}{f_{\mathrm{base}}} \qquad \ldots 3.3.1$$

where

t_{base} = nominal thickness of base material minus corrosion allowance, in millimetres

t_{clad} = nominal thickness of cladding material minus corrosion allowance, in millimetres

f_{clad} = design tensile strength for the cladding at the design temperature, in megapascals

f_{base} = design tensile strength for the base material at the design temperature, in megapascals

Where f_{clad} is greater than f_{base}, the multiplier f_{clad}/f_{base} shall be taken as equal to unity. Welded vessels in which the cladding is included in the computation of wall thickness shall be of Class 1 or 2A construction (see Table 1.6) when subject to internal pressure.

(d) *Composite tubes* Where tubes are made of composite materials and pressure and other loading conditions permit, the requirements of Clause 3.3.1.2(c) shall apply.

3.3.2 Reduced design tensile strength for low temperature service

Carbon and carbon-manganese steel pipe, plate, forgings, castings and welds may be used at temperatures down to 50°C below those permitted for design strength, in vessels where reduced pressures and reduced stresses occur at low operating temperatures, e.g. with liquefied gases in refrigeration vessels. Under these conditions the design tensile strength shall not exceed 50 MPa. (See Clause 2.6.)

NOTE: Where a vessel is subject to higher pressure at higher temperature, the design needs to also satisfy requirements for the higher pressure. Attention is particularly drawn to Clause 3.2.1.2.

3.3.3 Design compressive strength (f_c)

The design compressive strength for other than iron castings shall—

(a) not exceed the design tensile strength;

(b) comply with the requirements of Clause 3.7.5 for shells subject to axial compression; and

(c) comply with the requirements of Clause 3.9 for vessels subject to external pressure.

NOTE: Where buckling of components due to loads other than external pressure could occur, an analysis to determine safe working stresses should be undertaken by agreement between the parties concerned.

For iron castings where the design tensile strength is based on a factor of safety of 10 (see Table A1 of Appendix A), the design compressive strength shall not exceed twice the values given in Table B1(D).

3.3.4 Design shear strength (f_s)

Where shear stresses are present alone, the design shear strength shall not exceed 60% of the values given in Table B1, and shall not exceed 80% for restricted shear such as dowel bolts, rivets or similar construction in which the shearing member is so restricted that the section under consideration would fail without reduction in area.

3.3.5 Design bearing strength ($f_{bearing}$)

The design bearing strength shall not exceed 160% of the values given in Table B1.

3.3.6 Young's modulus (modulus of elasticity) (E)

The values for E are given in Table B3.

3.3.7 Design bending strength

The primary bending stress (f_b) across a solid section shall be limited to a value such that the total primary stress does not exceed 150 percent of the values given in Table B1.

NOTES:

1 This recognizes that bending stresses (and design peak stresses) can exceed the design tensile strength values of Table B1.

2 The equations in this Standard for various parts include provision for the above. However, when such stresses are to be calculated, the appropriate acceptance criteria are specified in Appendix H, using the design tensile strength 'f' listed in this Standard.

3 A solid section is defined as a solid plate and specifically excludes cross-sections such as a plate with stiffening ribs, a hollow section or the whole cross-section of a vessel.

3.4 THICKNESS OF VESSEL WALL

3.4.1 Minimum calculated thickness

The thickness obtained by the Clauses in Section 3 is that required to withstand the calculation pressure and where necessary shall be varied in accordance with Clause 3.4.2 and with provision made for any other design loadings given in Clause 3.2.3.

The dimensional symbols used in all design equations in this Section 3 represent dimensions in the corroded condition unless noted.

The nominal thickness so determined shall indicate the minimum class of construction required in accordance with Table 1.7. However, a higher class of construction may be used and appropriate credit taken. (See Clause 1.7 for other factors that require a higher vessel class.)

3.4.2 Thickness allowances

3.4.2.1 *Design thickness*

The actual thickness at any part of the completed vessel shall be no less than the design thickness, where the design thickness is equal to the minimum calculated thickness increased by the following allowances:

(a) Additional thickness for corrosion (see Clause 3.2.4).

(b) Additional thickness over that for pressure and corrosion considerations, sufficient to give necessary rigidity to permit handling and transport of the vessel and to maintain its shape at atmospheric or reduced pressure (see Clause 3.2.3).

The design thickness shall be not less than that required by Table 3.4.3.

Figure 3.4.2 shows the relationship between different thickness terms and thickness allowances.

3.4.2.2 *Further fabrication allowances*

When ordering material for fabrication of the vessel allowances shall be made to the design thickness to provide for the following:

(a) Except for plate material, additional thickness to allow for mill under-tolerance on the material (see appropriate material specification).

(b) For plate material, a thickness allowance to cater for mill under-tolerance on the material (see appropriate material specification). The ordered thickness minus the maximum specified mill under-tolerance shall be—

  (i) 0.94 of the design thickness, where design thickness is 5 mm or less; and

  (ii) the design thickness minus 0.3 mm, where design thickness is greater than 5 mm.

(c) Additional thickness to allow for reduction in thickness during fabrication operations, such as forming, machining and dressing of welds.

The nominal overall thickness for ordering (t_o, also denoted as t_s in some Clauses in this Standard) shall be as follows:

For material except plate

$$t_o = t + c + TA + MA + X \qquad \ldots 3.4.2(1)$$

For plate material where design thickness is 5 mm or less

$$t_o = 0.94(t + c) + TA + MA + X \qquad \ldots 3.4.2(2)$$

For plate material where design thickness is greater than 5 mm

$$t_o = t + c - 0.3 + TA + MA + X \qquad \ldots 3.4.2(3)$$

where

c = corrosion allowance, in millimetres

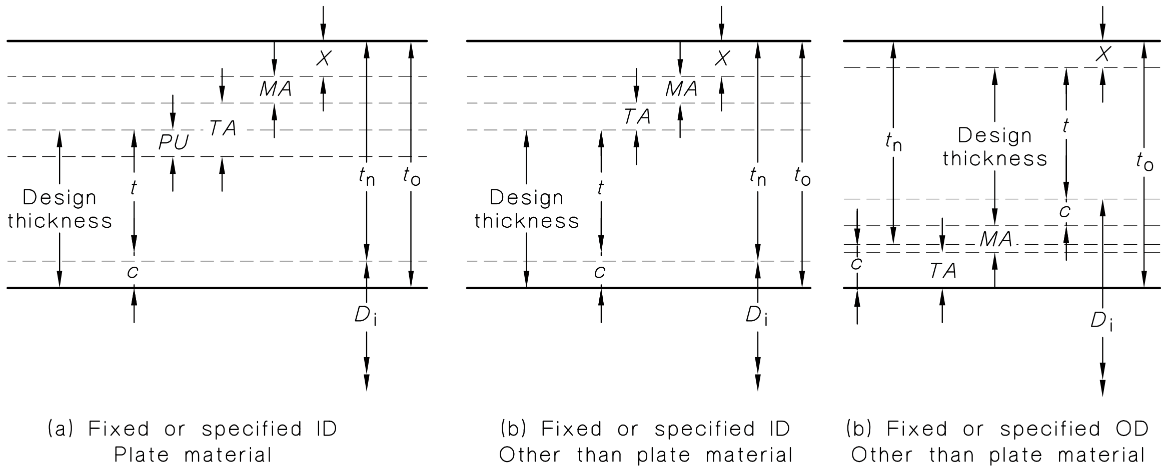
MA = manufacturing allowance (forming and machining), in millimetres

TA = undertolerance allowance (from the material specification) in millimetres

t = minimum calculated thickness for pressure and applied loads, in millimetres

X = any extra thickness required to round up to a commercially available thickness, in millimetres

Vessels made of plate complying with these provisions may be used at the design pressure appropriate to the above design thickness.



LEGEND:

t = Minimum calculated thickness for pressure and applied loads

c = Corrosion allowance

D_i = Internal diameter for calculation purposes

MA = Manufacturing allowance (forming and machining)

TA = Undertolerance allowance, from the material specification

PU = Permitted plate undertolerance (see Clause 3.4.2.2(ii), taken as 0 for all non-plate product forms (such as pipe, tube, forgings and castings)

X = Extra thickness to make up ordered thickness

t_o = Nominal overall thickness for ordering

t_n = Nominal thickness for use in various figures and for weld sizes.

Design thickness = $t + c$ (see Clause 3.4.2.1)

NOTE: This figure assumes corrosion occurs on the inside surface only. If not, the corrosion allowance shall be increased accordingly.

FIGURE 3.4.2 THICKNESS ALLOWANCES

3.4.3 Minimum nominal thickness of pressure parts

Notwithstanding the requirements of Clauses 3.4.1 and 3.4.2, the minimum nominal thickness of a pressure part shall comply with Table 3.4.3.

TABLE 3.4.3

MINIMUM NOMINAL THICKNESS OF ANY PRESSURE PART

Vessels constructed of metal	Outside diameter of vessel part (D_o) mm	Minimum nominal thickness for type of manufacture (see Notes 1 and 2): Forged; metal and submerged-arc welded mm	Brazed; GTAW welded; GMAW welded and heat exchanger tubes mm	Cast mm
All except as noted below (see Note 3)	≤225 >225 ≤1000 >1000	2.0 2.3 2.4	$0.10\sqrt{D_o}$ 1.5 2.4	4 8 10
Lethal contents	Twice the above values			
Transportable vessels	See Clause 3.26			
Vessel nozzles	See Clause 3.19.10.2			

NOTES:

1 Values are based primarily on limits of proven manufacture, assembly and ability to withstand handling, dispatch, installation and use.

2 Minimum thickness equals the total thickness for integrally clad vessels and the base thickness for applied linings.

3 Minimum thicknesses for group F and G steel parts are 5 and 6 mm respectively.

3.5 WELDED AND BRAZED JOINTS

3.5.1 Welded joints

3.5.1.1 *Types of welded joints*

For the purposes of this Standard, welded joints are classified as one of the following, according to their position as indicated in Figure 3.5.1.1 for typical joint types:

(a) *Type A, longitudinal* These are longitudinal welded joints in main cylindrical shells, transitions in diameter, or in nozzles or joints in positions requiring equivalent welds. This includes circumferential or any other welded joints within spherical shells, within formed or flat ends, or welds connecting spherical ends to main shells or within flat plates forming integral parts of pressure vessels.

Spiral welds are regarded as equivalent to longitudinal welds using $\eta = 1.0$, except as agreed between the parties concerned.

(b) *Type B, circumferential* These are circumferential welded joints within main cylindrical shells, within transitions in diameter, or within nozzles or circumferential welded joints connecting formed ends (other than spherical) or connecting transitions to main shells.

(c) *Type C, corner* These are peripheral welded joints with the weld located at a corner of a pressure-retaining part as in the joints connecting flanges, tube plates, or flat ends to main shells, to formed ends, to transitions in diameter, or to nozzles.

(d) *Type D, nozzle* These are welded joints connecting nozzles to main shells, to spheres, to transitions in diameter, or to ends.

In addition to the types of welded joints defined in AS 2812, butt joints are defined as—

(i) *double-welded butt joints,* i.e. butt joints welded from both sides; and

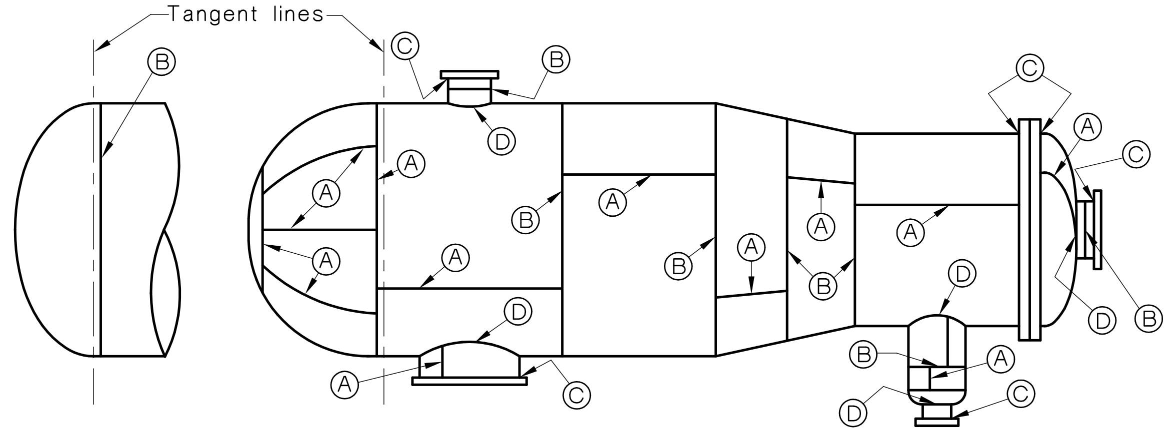
(ii) *single-welded butt joints*, i.e. butt joints welded from one side.

The following welded butt joints may be considered as double-welded butt joints:

(A) A single-welded butt joint where a backing strip is used and is subsequently removed and inspection shows complete penetration and fusion to the far side.

(B) A single-welded butt joint using a process and procedure so that subsequent inspection shows complete penetration and fusion to the far side, including welds using temporary backing bars.

(C) Electroslag, electrogas, flash butt, resistance, and similar welds.



NOTE: For explanation of points A to D, see Clause 3.5.1.1

FIGURE 3.5.1.1 WELDED JOINT TYPES—BASED ON LOCATION

3.5.1.2 *Number of joints*

The number of welded joints in a vessel shall be the minimum practicable number.

3.5.1.3 *Location of joints*

Requirements for location of welded joints shall be as set out in AS 4458.

Welded joints should be located as follows:

(a) To avoid disturbances to the flow of force or sudden changes in stiffness or areas of severe stress concentration particularly in vessels subject to fluctuating or impact loads. See also Clause 3.18.5.3 concerning openings.

(b) Clear of areas of severe corrosion.

(c) To avoid more than two welded joints meeting at a point.

(d) So the distance between the toes of attachment welds, the toes of large fillet welds of nozzles or undressed main welds is not less than the smaller of 40 mm or three times the shell thickness.

(e) To provide reasonable access for welding equipment and welder and for visual, radiographic or ultrasonic examination of the root side of butt welds.

(f) So the joint is readily visible in service (after removal of insulation, if necessary) and is clear of supports, internals and their attachments.

3.5.1.4 *Design of welded joints*

3.5.1.4.1 *General*

The types of welded joints shall be adequate to transfer all expected loads between parts joined.

Edge preparation of welds shall enable sound welds and complete fusion and penetration to be obtained consistently with the particular welding procedure.

3.5.1.4.2 *Butt welds*

The throat thickness (excluding any weld reinforcement or weld metal extending outside the projection of the parent material) of longitudinal and circumferential type butt welds in shells, ends or nozzles, shall be at least equal to the thickness of the thinner of the parts joined.

3.5.1.4.3 *Fillet welds*

Fillet welded circumferential type joints are not permitted, except as described in Figure 3.5.1.5(A) and Clause 3.5.1.7, where the dimensions shall develop the strength required for the appropriate joint efficiency (see Clause 3.5.1.7).

The allowable load on fillet welds at nozzle connections shall be in accordance with Clause 3.19.3.5.

The allowable load on other fillet welds shall be based on the minimum throat area of the weld and using a design strength not more than 75% of the design strength, f, for the weaker material in the joint.

The minimum design throat area shall be taken as the design throat thickness allowing for the reduction in the throat thickness made necessary by any root gap, multiplied by the effective weld length, which equals the length measured on the centre-line of the throat. No fillet weld shall have an effective weld length less than 50 mm or six times the leg length, whichever is less.

The shape of fillet welds shall be in accordance with Figure 3.5.1.4.

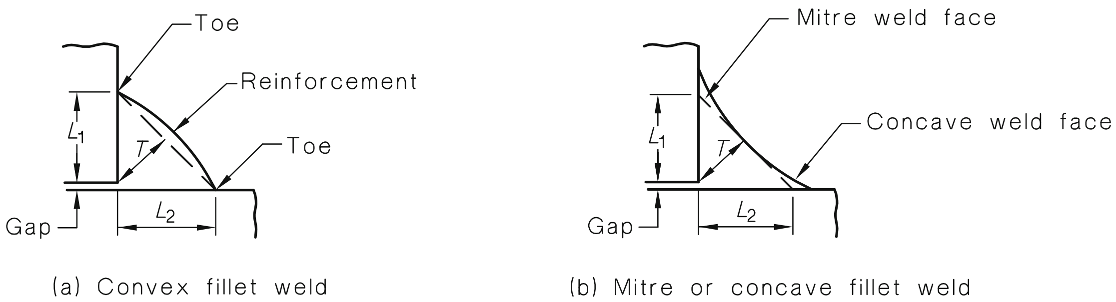
For fillet welds in corner or nozzle welded joints, and other joints subjected to bending stresses, see Clause 3.5.1.4.5.

The plates of fillet welded lap joints shall be lapped at least four times the thickness of the thinner plate, except for lap-welded dished ends (see Clause 3.12.6).

3.5.1.4.4 *Plugwelds and slotwelds*

Plugwelds and slotwelds shall be used only where other methods of welding attachment are not possible to achieve the required joint efficiency of lap joints, and in reinforcements around openings and in non-pressure structural attachments. Except for stayed surfaces (see Clause 3.16), plugwelds or slotwelds shall not be considered to take more than 30% of the total load to be transmitted.

Where holes or slots in one or more of the parts forming the joint are manually welded, the hole or slot shall not be filled with weld metal, nor partially filled in such a manner as to form a direct weld metal connection between opposite sides of the hole. The diameter of the hole or width of the slot shall be no less than 2.5 times the thickness of the plate in which the hole is made. The ends of slots shall be semicircular or rounded with a radius no less than 1.25 times the plate thickness.



LEGEND:

L_1 = Effective leg length on vertical face
L_2 = Effective leg length on horizontal face
T = Design throat thickness ($0.71L_1$ for equal leg fillet)
Gap = 1.5 mm or $L_1/8$, whichever is less
Reinforcement: minimum = 0
Reinforcement maximum: = 1.5 mm + $L_1/8$, or 4mm, whichever is less

FIGURE 3.5.1.4 FILLET WELD SHAPE AND DIMENSIONS

Where automatic or semi-automatic processes are used for making plug welds, a hole smaller than required for manual welding may be adopted and the plug hole completely filled with weld metal, provided the manufacturer proves by procedure tests that complete fusion and penetration can be obtained consistently and the quality of the welding complies with requirements of this Standard.

The distance from the edge of the plate or member and the edge of the hole or slot shall be no less than twice the thickness of the plate or member.

The strength of ligaments between plugwelds and slotwelds shall be no less than 50% of the solid plate. The strength of plugwelds and slotwelds shall be calculated in accordance with Clause 3.5.1.4.3.

3.5.1.4.5 *Welded joints subject to bending stresses*

If welded joints are subjected to bending stresses then fillet welds shall be added where necessary to reduce stress concentration.

Corner or T-joints may be made with fillet welds only, provided the plates forming the joint are properly supported independently of such welds except where specific weld details are permitted in other Clauses of this Standard and AS 4458. However, independent supports are not required for joints such as lugs for platforms, ladders and other attachments.

3.5.1.4.6 *Welded joints with backing strip*

For limitations see Table 3.5.1.7.

3.5.1.4.7 *Corner and nozzle welded joints*

For design of corner and nozzle welded joints see Clauses 3.15 and 3.19, respectively.

3.5.1.4.8 *Stud welds*

Stud welds shall not be used for connecting pressure-retaining parts.

3.5.1.5 *Acceptable joint preparation*

Some acceptable types of joint preparation for joints within shells and ends are given in Figures 3.5.1.5(A) to (E). For acceptable types of joints for attachments of flat ends, nozzles, and the like, see appropriate clauses for these components.

Where pressure welding processes are used, butt type joints only are permitted.

Where joint preparations other than those shown in this Standard are required, these shall be proven by qualification of the welding procedure in accordance with AS/NZS 3992.

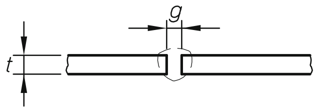
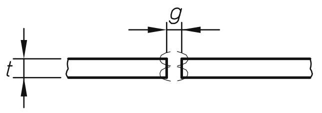
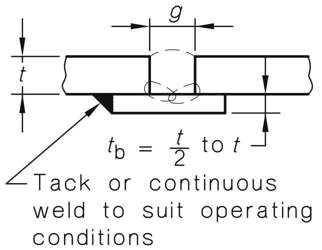
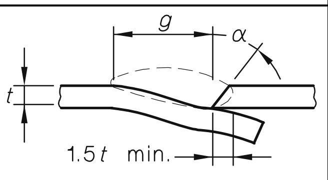
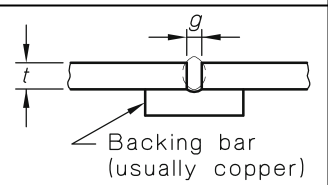
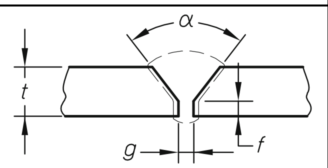
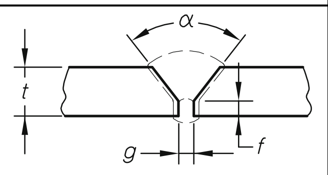
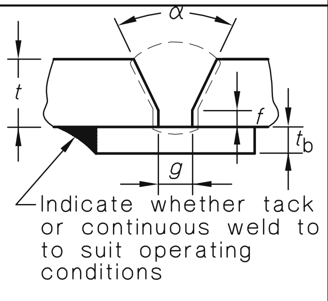
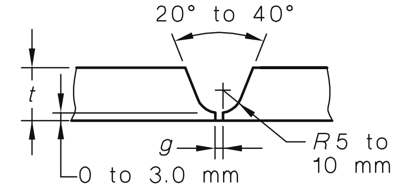
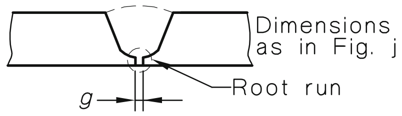
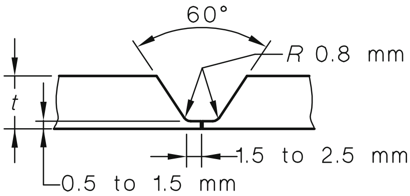
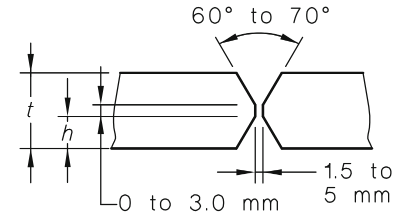
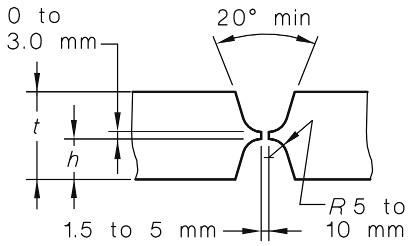
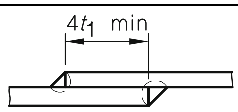
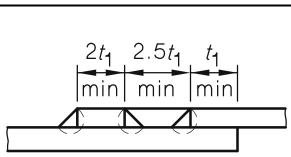
Figure	Joint type (Note 1)	Joint form (sectional view) (Note 2)	Recommended thickness (*t*) mm	Dimensions of joint: Gap (*g*) mm	Bevel angle (α)	Root face (*f*) mm	Application and Notes (See also joint limits in Table 3.5.1.7)
a	Single-welded square butt joint		1.5 3 max.	0–1.5 0–2.5	— —	— —	Circumferential joints but not recommended.
b	Double-welded square butt joint		1.5 3 5 max.*	0–1.5 0–2.5 0–3	— — —	— — —	Circumferential and longitudinal joints. *To 10 mm with deep penetration welding procedures.
c	Single-welded square butt joint with backing strip		3 5 6 max.	3–6 5–8 6–10	— — —	— — —	Figure (c) may be used for longitudinal joints where one side is inaccessible for welding. Figures (c) and (d) may be used for circumferential joints where one side is inaccessible for welding, and corrosion or fatigue is not important.
d	Single-welded joggled butt		16 max.	*t*–2.5t	0–30°	—	Close fit of backing strip, joggle and backing bar is essential. Where backing strip or joggle is machined out after welding this weld is suitable for longitudinal joints provided the root is suitably examined.
e	Single-welded square butt joint with backing bar		≤ 5	*t*/2 max	—	—	Circumferential and longitudinal joints, provided the root is suitably examined, and corrosion and fatigue are not important.
f	Single-welded single V butt joint (Note 3)		3–10; over 10	1.5–3 1.5–5	60°–70° 60°–70°	0–1.5 0–3	Circumferential joints where one side is inaccessible for welding, and corrosion or fatigue is not important. Larger angles may be used for vertical welds
g	Double-welded single V butt joint (Note 3)		All	0–3	60°–70°	0–3	Circumferential and longitudinal joints. Second side grooved to sound metal before welding second side. The V should be located on the inside of small diameter vessels.

Figure	Joint type (Note 1)	Joint form (sectional view) (Note 2)	Recommended thickness (*t*) mm	Min gap for α 45°	Min gap for α 30°	Min gap for α 15°	*f*	t_b	Application and Notes
h	Single-welded single V butt joint with backing strip (Note 3)		5 6 10 12 Over 12 Over 25	5 5 6 8 10 11	6 6 8 10 10 11	8 8 10 11 11 12	0–1.5 0–1.5 0–1.5 0–3 0–3 0–5	2.5–5 3–6 3–8 3–10 3–*t*/2 5–*t*/2	Circumferential and longitudinal joints where one side is inaccessible for welding and corrosion or fatigue is not important. Longitudinal joints are limited to 16 mm max. Where the backing strip is machined out after welding, this weld is suitable for all longitudinal welds provided the root is suitably examined.

NOTE: See end of Table 3.5.1(A) for Notes.

FIGURE 3.5.1.5(A) (in part) SOME TYPICAL WELD PREPARATIONS—CARBON, CARBON-MANGANESE, ALLOY AND AUSTENITIC CHROMIUM-NICKEL STEELS—MANUAL AND GAS METAL ARC WELDING PROCESSES
(Suitable for all positions of welding, but downhand preferred)

Figure	Joint type (Note 1)	Joint form (sectional view) (Note 2)	Recommended thickness (t) mm	Application and Notes (See also joint limits in Table 3.5.1.7)
j	Single-welded single U butt joint		15 max.	Circumferential welds, where one side is inaccessible for welding g = 0–3 mm
k	Double-welded single U butt joint		Over 15 to 25	Longitudinal and circumferential welds g = 0–5 mm
l	Single-welded single U butt joint (GTAW root)		20 max.	Circumferential welds where inside is inaccessible for welding. Root run is to be made by GTAW process with inert gas backing
m	Double-welded double V butt joint		15–38	Longitudinal and circumferential welds h may vary from $\frac{t}{2}$ to $\frac{t}{3}$
n	Double-welded double U butt joint		over 25	Longitudinal and circumferential welds h may vary from $\frac{t}{2}$ to $\frac{t}{3}$
p	Double full fillet lap joint		10 max. 12 max.	Longitudinal welds in Class 3 vessels Circumferential welds in Class 3 vessels t_1 = thickness of thinner plate
q	Single full fillet lap joint with plug welds		12 max.	Circumferential welds in Class 3 vessels for attachment of ends to shells 610 mm max. inside diameter t_1 = thickness of thinner plate Plug welds are to be proportioned to take 30% of total load

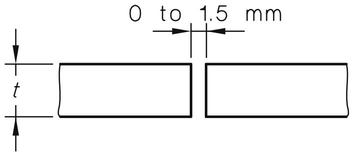
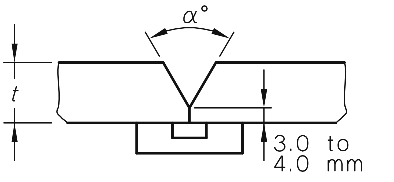
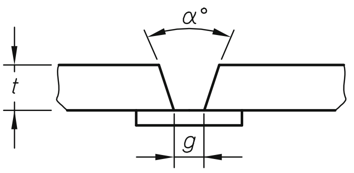
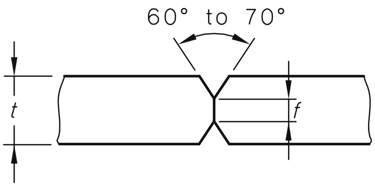
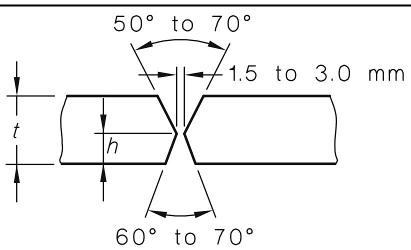
NOTES:

1 For austenitic steels, (f), (g) and (j) to (n) are recommended.

2 The use of minimum angle should be associated with maximum radius or gap. Conversely, the minimum radius or gap should be associated with the maximum angle.

3 Alternatively, in lieu of (f), (g) or (h), single bevel preparation in accordance with Figure 3.19.3(D) may be used.

FIGURE 3.5.1.5(A) (in part) SOME TYPICAL WELD PREPARATIONS—CARBON, CARBON-MANGANESE, ALLOY AND AUSTENITIC CHROMIUM-NICKEL STEELS—MANUAL AND GAS METAL ARC WELDING PROCESSES
(Suitable for all positions of welding, but downhand preferred)

Figure	Joint type	Joint form (sectional view) (see Note)	Recommended thickness (t) mm	Application and Notes (See also joint limits in Table 3.5.1.7)
a	Single-welded square butt joint (with temporary backing)		1.5 to 8	Temporary backing bar required
b	Double-welded square butt joint		3 to 12	Second side need not be cut back to sound metal if the root runs penetrate each other
c	Single-welded single V butt joint (with temporary backing)	 t mm: 5, 12, 20, 25, 38 α°: 60, 60, 45, 35, 30	5 to 38	Longitudinal and circumferential welds. Temporary backing bar may be copper or flux covered.
d	Single-welded single V joint (with backing strip)	 t mm / α° / g 5 to 20 / 30 to 50 / 0 to 3 over 20 / 20 to 30 / 10 to 15*	5 and over	Welded in several layers with backing strip. Where the backing strip is retained after welding (see Clause 3.5.1.6), same limits as in Figure 3.5.1.5(A)(h) apply *To 25 mm only where contraction assists in imparting shape required
e	Double-welded double V butt joint	 t mm: ≤12, ≤25, 38, 50, 63 f mm: 6, 8, 11, 12, 15	10 and over	Longitudinal and circumferential welds. Second side need not be cut back to sound metal if the root runs penetrate each other Root-face may be off-centre Gap: 0 to 1.5 mm
f	Double-welded double V butt joint (manual backing)		10 and over	Manual metal arc weld may be laid and cut back before submerged arc welding. h = 5 mm for t <12 mm = 6 mm min. for t ≥12 mm

NOTE: The use of minimum angle should be associated with maximum gap. Conversely, the minimum gap should be associated with the maximum angle.

FIGURE 3.5.1.5(B) SOME TYPICAL WELD PREPARATIONS—CARBON, CARBON-MANGANESE, ALLOY AND AUSTENITIC CHROMIUM STEELS—SUBMERGED ARC WELDING PROCESS

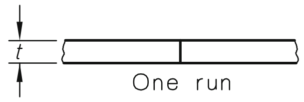
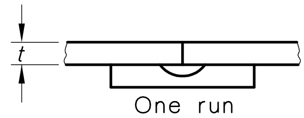
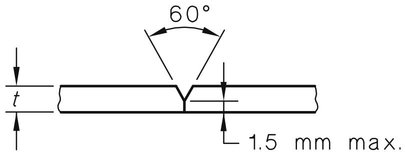
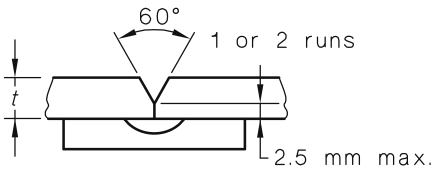
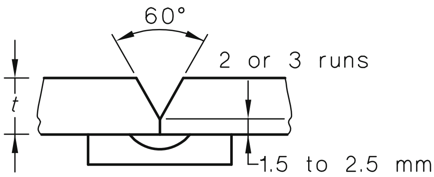
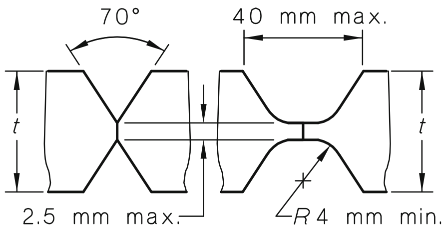
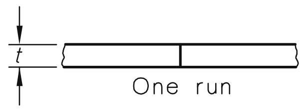
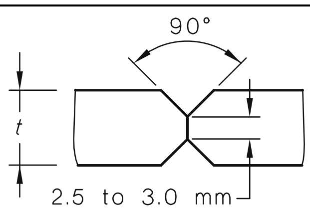
Figure	Joint type	Joint form (sectional view)	Recommended thickness, (t) mm	Application and Notes (See also joint limits in Table 3.5.1.7)
a	Single-welded square butt joint		3 max.	Inert gas backing or backing bar may be used
b	Single-welded square butt joint, with backing bar		3 max.	Backing bar required
c	Single-welded single V joint		3 and over	Either a backing bar or argon backing should be used. There should be no access for air to the back of the weld
d	Single-welded single V joint, with backing bar		5 max.	Frequently a filler rod is not used for the first run. Where the back of the joint cannot be dressed after welding, argon backing should be used, and there should be no access for air to the back of the weld
e	Single-welded single V joint, with backing bar (or with sealing run, i.e. double-welded)		7 max.	Where no backing bar is used, cut back to sound metal and add sealing run
f	Double-welded double V joint		6 and over	Cut back after first run to sound metal before welding underside
g	Single-welded square butt joint		3 max.	Butt welds in plate not exceeding 3 mm thick Double operator single run vertical GTAW process
h	Double-welded double V joint		3 to 6	Butt welds in plate between 3 mm and 6 mm thick Double operator single run vertical GTAW process

FIGURE 3.5.1.5(C) SOME TYPICAL WELD PREPARATIONS—AUSTENITIC CHROMIUM-NICKEL STEELS—GMAW AND GTAW PROCESSES

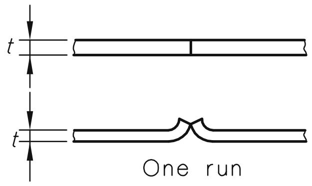
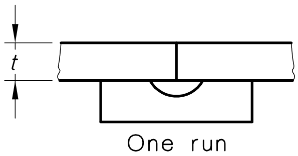
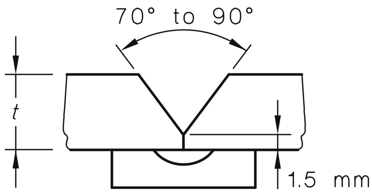
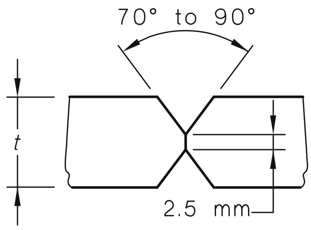
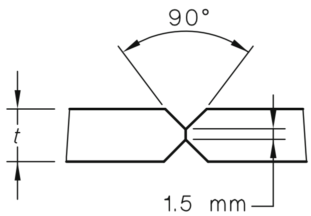
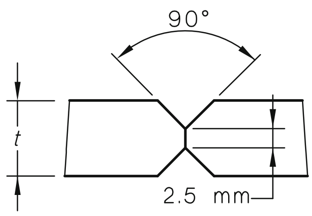
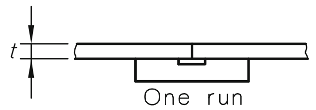
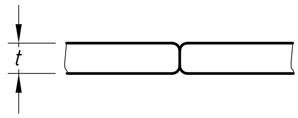
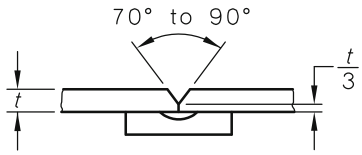
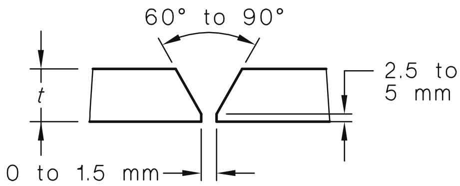
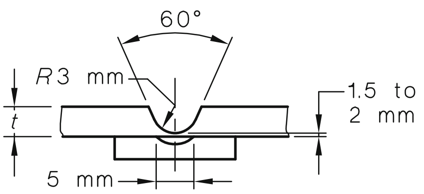
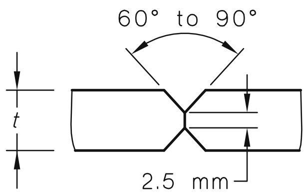
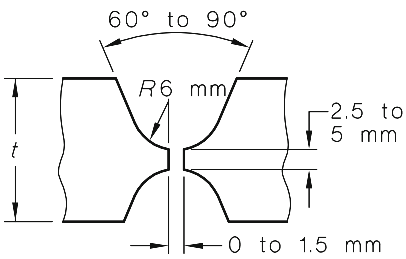
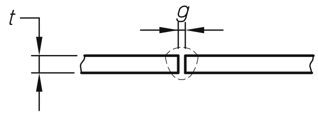
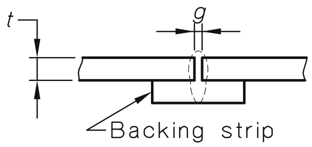
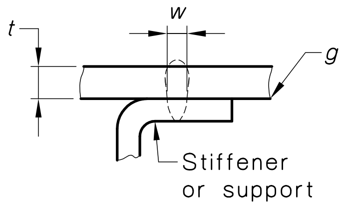
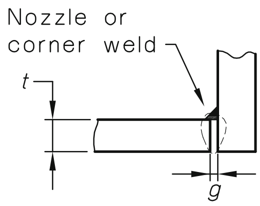
Figure	Joint type	Joint form (sectional view)	Recommended thickness (t) mm	Application and Notes (See also joint limits in Table 3.5.1.7)
a	Single-welded square or flanged butt joint		2 max. 1.5 max.	Flanging would be used only where square-edge close butt welds are impracticable. If backing bar is used it should conform to Figure 3.5.1.5(E)(a)
b	Single-welded square butt joint with backing bar		2 to 5	Where a backing bar cannot be used, welding from both sides is recommended
c	Single-welded single V butt joint with backing bar (or with sealing run, i.e. double-welded)		6 to 10	Where no backing bar is used, it is good practice to chip back to sound metal and add sealing run
d	Double-welded double V butt joint		5 to 12	Chip back first run to sound metal before welding underside. Preheating may be necessary
e	Double-welded double V butt joint		5 to 6	Vertical butt welds by the double operator technique
f	Double-welded double V butt joint		6 to 12	

FIGURE 3.5.1.5(D) SOME TYPICAL WELD PREPARATIONS—ALUMINIUM AND ALUMINIUM ALLOYS—GTAW PROCESS

Figure	Joint type	Joint form (sectional view) (see Note)	Recommended thickness (t) mm	Application and Notes (See also joint limits in Table 3.5.1.7)
a	Single-welded square butt with backing bar		1.5 to 5	—
b	Double-welded square butt		6 to 10	Weld from both sides, sighting Vs recommended. 6 mm is maximum material thickness for positional welding
c	Single-welded single V butt joint with backing bar		5 to 12	Weld in one run. Suitable also for positional welding, when welded from both sides
d	Single-welded single V butt joint		6 to 12	One or more runs from each side. Back chipping recommended after first run
e	Single-welded single U butt joint with backing bar		6 to 20	One or more runs from one side, depending on thickness. Suitable also for position welding
f	Double-welded double V butt joint		12 to 25	Up to 1.5 mm root gap. One or more runs from each side. Back chipping recommended after first run
g	Double-welded double U butt joint		12 to 25	—

NOTE: The use of minimum angle should be associated with maximum radius or gap. Conversely, the minimum radius or gap should be associated with the maximum angle.

FIGURE 3.5.1.5(E) SOME TYPICAL WELD PREPARATIONS—ALUMINIUM AND ALUMINIUM ALLOYS—GMAW PROCESS

Figure	Joint type	Joint form (sectional view) (Note 2)	Recommended thickness (t) mm	Gap (Note 1) (g) mm	Application and Notes (See also joint limits in Table 3.5.1.7)
a	Single-welded square butt joint		0.5 to 20 max.	Note 1	Circumferential joints and longitudinal joints provided the root is suitably examined (Note 3)
b	Single-welded square butt joint with backing strip		0.5 to 20 max.	Note 1	Circumferential and longitudinal joints where one side is inaccessible for welding, and corrosion and fatigue is not important. Close fit of the backing strip is essential.
c	Single-welded weld through with backing strip		0.5 to 20 max.	0-0.2 Note 4	One sided welding of lap type joints, e.g. stiffeners, supports, or dimpled plates (before or after dimpling). Not for corrosive of fatigue service. (Note 3)
d	Single- (or double) welded tee or corner joint		0.5 to 5 max.	Note 1	Joints where one or two sides are accessible provided the root is examined. (Note 3)

NOTES:

1 g = 0 to 0.5 mm, or 0.25 × focussed beam width

2 Filler metal is recommended, however, where no filler metal is added, joint corners should be sharp, not radiused to minimize shrinkage. Slight underfill may be acceptable, or where t has added allowance.

3 When minimum distortion is essential.

4 w = width of weld

Width of weld should be adequate for the required load.

FIGURE 3.5.1.5(F) SOME TYPICAL WELD PREPARATIONS—LASER BEAM WELDING PROCESS

(Suitable for all positions of welding, but downhand preferred)

3.5.1.6 *Application of welded joints*

The application of various types of welded joints longitudinal and circumferential type joints shall be in accordance with Table 3.5.1.7.

Butt-welded joints using backing strips retained in service or single-welded lap joints shall not be used where excessive corrosion or fatigue due to fluctuating or impact loads are likely to occur.

For Group G steels, welded joint Types A or B shall be double-welded butt joints or other butt joints with equivalent quality, and for joint Type C shall be full penetration welds extending through the entire section at the joint without retained backing strip.

For Group F steels, welded joint Types A, B and C shall be double-welded butt joints or other joints with equivalent quality except that for circumferential welds a single-welded butt joint with retained backing strip may be used.

3.5.1.7 *Welded joint efficiency, η*

The maximum allowable joint efficiency of welded joints shall be in accordance with Table 3.5.1.7.

TABLE 3.5.1.7

WELDED JOINT EFFICIENCIES (See Note 5)

Type of joint	Permissible joint location (See Figure 3.5.1.1)	Joint limitations (Notes 3 & 6)	Radiographic or ultrasonic examination (Note 1)	Maximum welded joint efficiency for vessel (Note 4)			
				Class 1 1H, 2H S, 2S	Class 2A	Class 2B	Class 3
Double-welded butt joint, or other butt joint with equivalent quality (welds using backing strips that remain in service are excluded)	A,B,C,D	None	Full Spot None	1.0 1.0 (2H,2S) —	— 0.85 —	— — 0.80	— — 0.70
Single-welded butt joint with backing strip that remains in service	A,B,C,D	Circumferential joints—no limit except; $t \leq 16$ mm for joggled joint (See Figure 3.5.1.5(A)(d) Longitudinal joints-limited to $t \leq 16$ mm	Full Spot None	1.0 — —	— 0.80 —	— — 0.75	— — 0.65
Single-welded butt joint without use of backing strip	B,C	Circumferential joints only in Class 2 and Class 3 vessels with $t \leq 16$ mm and 610 mm max. inside diameter	None	—	0.70	0.65	0.6
Double full fillet-welded lap joint. Figure 3.5.1.5(A)(p) & 3.12.6(g)	A,B,C	Circumferential joints in Class 3 vessels only. Longitudinal joints in Class 3 vessels only with $t \leq 10$ mm	None	—	—	—	0.55
Single full fillet lap joint with plug welds conforming to Figure 3.5.1.5(A)(q)	B	Circumferential joints only in Class 3 vessels for the attachment of dished ends to shells 610 mm max. inside diameter (Note 2)	None	—	—	—	0.50
Single full fillet-welded lap joint without plug welds conforming to Figure 3.12.6(h), (j) and (l)	B	Circumferential joints only in Class 3 vessels for the attachment of— (a) ends convex to pressure, to shells with fillet weld on inside of shell with $t \leq 16$ mm (b) ends concave to pressure, to shells with $t \leq 8$ mm thickness 610 mm max inside diameter with fillet weld on end flange only	None	—	—	—	0.45
Welded joints in pipes and tubes	A,B	For longitudinal welds in high alloy steel pipes, the joint efficiencies have been included in the design strength listed in Table B1(B). For carbon, carbon-manganese and alloy steel pipes, the joint efficiencies for longitudinal welds as specified in AS 4041 shall be used.					

NOTES TO TABLE 3.5.1.7:

1 The examination listed is for Type A and B joints. See AS 4037 for examination of all types of joints.

2 See Clause 3.23 for exceptions for some jacketed vessels.

3 See Clause 3.5.1.6 for requirements for specific materials.

4 These efficiencies apply to longitudinal and circumferential type welds (see Clause 3.5.1.1).

5 A welded joint efficiency of 1.0 shall be applicable for design purposes for—

(a) seamless products, such as seamless pipes and forgings; and

(b) longitudinal and circumferential type butt welds and fillet welds attaching ends, on vessels subject to vacuum only.

5 t = nominal thickness of shell.

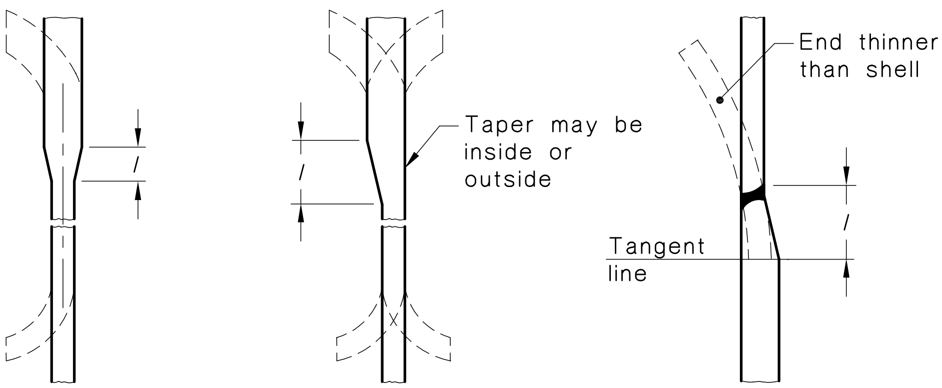
3.5.1.8 *Butt welding between plates of unequal thickness*

Where two plates to be welded by a butt joint differ in thickness by more than 25% of the thinner plate, or by more than 3 mm, the thicker plate shall be reduced at the abutting edge on either the inside or the outside or both, as shown in Figure 3.5.1.8. In all such cases, the edge of the thicker plate shall be trimmed to a smooth taper extending for a distance of at least three times (or four times for Class 1H and 2H vessels) the offset between the abutting surfaces, so that the adjoining edges will be approximately the same thickness. The length of the required taper may include the width of the weld.

For plates using a double-welded double vee preparation the difference between the surfaces of both plates may be not more than 3 mm on each side before tapering of the thicker plate is required.

When the weld is required to be radiographically examined, the maximum thickness through the weld shall comply with AS 4037.

For attachment of ends to shells of differing thicknesses see Clause 3.12.6.



(a) Preferred (centrelines coincide) (b) Permissible (c) Preferred

NOTES:

1 In all cases except for Class 1H and 2H vessels, *l* should be no less than three times the offset between the abutting plates. For Class 1H and 2H vessels, *l* shall be no less than four times the offset.

2 Length of required taper, *l*, may include the width of the weld.

3 Misalignment of centre-lines ≤ ½ (*t* thick − *t* thinner)

FIGURE 3.5.1.8 BUTT WELDING BETWEEN PLATES OF UNEQUAL THICKNESS

3.5.2 Riveted joints

Riveted joints shall not be permitted for Class 1H and 2H vessels.

3.5.3 Brazed joints

3.5.3.1 *General*

The following requirements apply specifically to pressure vessels and parts thereof that are fabricated from suitable materials listed in Table B1 by brazing in accordance with the general requirements of this Standard.

Brazed joints shall not be used for the following:

(a) Vessels with lethal contents (as per AS 4343).

(b) Directly fired vessels.

(c) Joints at design temperatures above 95°C, except brazing filler metal B-CuP may be used up to 105°C maximum and B-Ag, B-CuZn, B-Cu and B-Al-Si may be used up to 205°C maximum provided joint tensile test shows a tensile strength and yield strength not less than the minimum tensile and yield strength of the weaker of the parent metals at the design temperature. If the design is based on creep properties, the joint creep strength shall be similarly proven.

(d) Class 1H and 2H vessels.

3.5.3.2 *Strength of brazed joints*

The designer is responsible to determine from suitable tests or from past experience that the specific brazing filler metal selected can produce a joint that will have adequate strength over the operating temperature range. AS 4458 specifies details for qualification requirements.

The strength of the brazed joint shall be no less than the strength of the parent material, or the weaker of two parent materials in case of dissimilar metal joints, for all temperatures within the operating range.

3.5.3.3 *Corrosion allowance*

Provision shall be made for corrosion in accordance with the requirements of Clause 3.2.4.

Corrosion of the brazing filler metal and galvanic action between the brazing filler metal and the base material shall be considered in selecting the brazing filler metal.

The plate thickness in excess of that calculated for a seamless vessel taking into account the applicable loadings in Clause 3.2.3 may be taken as an allowance for corrosion in vessels that have longitudinal joints of double strap butt-joint type. Additional corrosion allowance shall be provided when needed, particularly on the inner buttstraps.

The requirements of this Standard are not intended to apply to brazing used for the attachment of linings of corrosion-resistant material that are not counted on to carry load but resultant galvanic action shall still be considered.

3.5.3.4 *Brazed joint efficiency*

The brazed joint efficiency to be used in the design of pressure vessels and parts thereof shall be 1.0 for joints in which visual examination shows that the brazing filler metal has penetrated the entire joint (see Figure 3.5.3.4(a)).

The brazed joint efficiency to be used in the design of pressure vessels and parts thereof shall be 0.5 for joints in which visual examination will not provide proof the brazing filler metal has penetrated the entire joint (see Figure 3.5.3.4(b)).

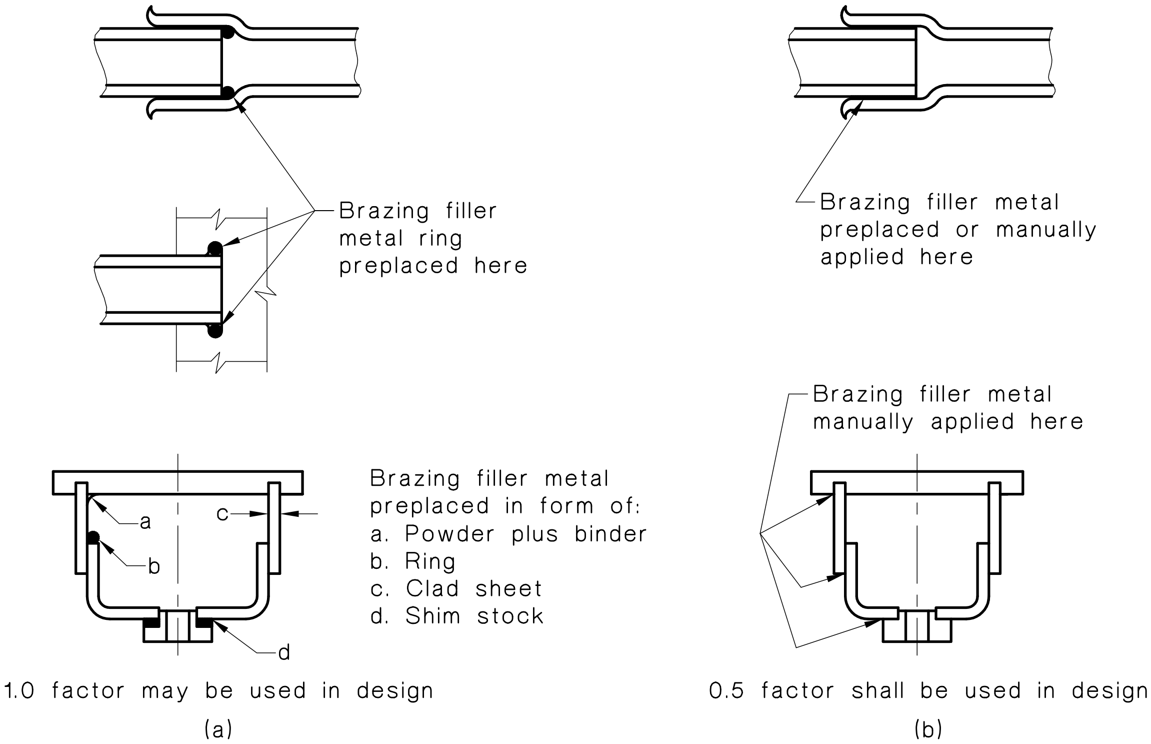


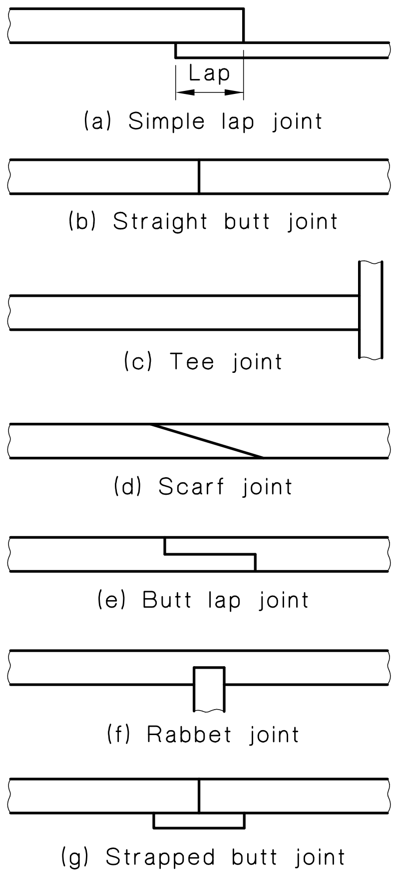
FIGURE 3.5.3.4 EXAMPLES OF FILLER METAL APPLICATION

3.5.3.5 *Application of brazing filler metal*

The design shall provide for the application of the brazing filler metal as part of the design of the joint. Where practicable, the brazing filler metal shall be applied in such a manner that it will flow into the joint or be distributed across the joint and produce visible evidence that it has penetrated the joint.

3.5.3.6 *Permissible types of joints*

Some permissible types of brazed joint are shown in Figure 3.5.3.6. For any type of joint, the strength of the brazed section shall exceed that of the base material portion of the test specimen in the qualification tension tests provided for in AS/NZS 3992. Lap joints shall have an overlap of five times the thickness of the thinner plate for longitudinal joints and not less than three times the thickness of the thinner plate for circumferential joints to provide a higher strength in the brazed joint than in the base material.



NOTE: Other equivalent geometries yielding substantially equal results also are acceptable.

FIGURE 3.5.3.6 SOME ACCEPTABLE TYPES OF BRAZED JOINTS

3.5.3.7 *Joint clearance*

The joint clearance shall be kept sufficiently small so the filler metal will be distributed by capillary action and shall be within the tolerances specified in the joint design and in the qualified brazing procedure (see Table 3.5.3.7).

TABLE 3.5.3.7

RECOMMENDED JOINT CLEARANCES AT BRAZING TEMPERATURE

Brazing filler metal classification (ANSI/AWS A5.8)	Clearance mm
B-Al-Si group	0.15–0.25 for laps less than 6 mm
	0.25–0.64 for laps greater than 6 mm
B-Cu-P group	0.03–0.13
B-Ag group	0.05–0.13
B-Cu-Zn group	0.05–0.13
B-Cu-group	0.0–0.05

3.5.4 Soldered joints

Soldered joints are permitted for small vessels or parts provided the following are complied with:

(a) Vessel contents are not lethal.

(b) The vessel is not directly fired.

(c) The design temperature does not exceed 50°C.

(d) The joints comply with requirements equivalent to those in Clause 3.5.3.

(e) The joints are shown to be suitable for the particular application.

3.6 LIGAMENT EFFICIENCY

Where a cylindrical shell is drilled with multiple holes, the ligament efficiency (η) to be used in determining the thickness of the shell shall be determined in accordance with AS 1228 or other method agreed by the parties concerned.

For ligament efficiency in unstayed flat tubeplates, see Clause 3.17.1.

3.7 CYLINDRICAL AND SPHERICAL SHELLS SUBJECT TO INTERNAL PRESSURE AND COMBINED LOADINGS

3.7.1 General

The minimum thickness of cylindrical or spherical shells or cylindrical or spherical parts of vessels subject to internal pressure and, when applicable, combined loading, shall be no less than that determined in accordance with this Clause (3.7) and Clause 3.4.3. See also Clause 3.8.

3.7.2 Notation

For the purpose of Clauses 3.7 and 3.8 the following notation applies:

D = inside diameter of shell, in millimetres

D_m = $\frac{D+D_o}{2}$ = mean diameter of shell, in millimetres

D_o = outside diameter of shell, in millimetres

E = Young's modulus at design temperature, in megapascals

f = design tensile strength at the design temperature (see Clause 3.3), in megapascals

f_a = f at test temperature, in megapascals

M = longitudinal bending moment, in newton millimetres

P, P_h = calculation pressure P (see Clause 3.2.1), or the pressure under hydrostatic test P_h, as appropriate, in megapascals

Q = torque about vessel axis, in newton millimetres

S_b = bending stress in vessel, in megapascals

S_c = circumferential pressure stress in vessel, in megapascals

S_E = equivalent stress at any given point in vessel (maximum shear stress basis), in megapascals

S_l = longitudinal stress in vessel, in megapascals

S_s = shear stress in vessel, in megapascals

S_w = weight stress in vessel, in megapascals

t = minimum calculated thickness of the pressure part (exclusive of added allowances, see Clause 3.4.2), in millimetres

T = actual thickness (taken as nominal thickness less allowances), in millimetres.

W = (vertical vessel only)—

(a) for points above plane of support: force due to the mass of the vessel, fittings, attachments and fluid supported above point considered, the sum to be given a negative sign in Equation 3.7.5(1), in newtons; and

(b) for points below plane of support: force due to the mass of the vessel, fittings and attachments below point considered, plus fluid content, the sum to be given a positive sign in Equation 3.7.5(1), in newtons

α = half apex angle of a conical vessel shell or skirt, in degrees (for cylindrical sections, taken as = 0)

η = efficiency of the welded joint or any line of holes or ligaments in the shell, whichever is the least (see Clauses 3.5 and 3.6)

3.7.3 Cylindrical shells

The minimum calculated thickness of a cylindrical shell shall be the greater thickness determined from the following equations:

(a) Based on circumferential stress (longitudinal joints)—

$$t = \frac{PD}{2f\eta - P} = \frac{PD_{\mathrm{m}}}{2f\eta} = \frac{PD_{\mathrm{o}}}{2f\eta + P} \qquad \ldots 3.7.3(1)$$

(b) Based on longitudinal stress (circumferential joints)—

$$t = \frac{PD}{4f\eta - P} = \frac{PD_{\mathrm{m}}}{4f\eta} = \frac{PD_{\mathrm{o}}}{4f\eta + P} \qquad \ldots 3.7.3(2)$$

3.7.4 Spherical shells

The minimum calculated thickness of a spherical shell shall be determined from the following equation:

$$t = \frac{PD}{4f\eta - P} = \frac{PD_{\mathrm{m}}}{4f\eta} = \frac{PD_{\mathrm{o}}}{4f\eta + P} \qquad \ldots 3.7.4$$

NOTE: Equation 3.7.4 is based on burst considerations rather than the Lame equation, and so differs slightly from previous editions of this Standard. This requirement is not intended to apply retrospectively (see AS/NZS 1200, Clause 1.12).

3.7.5 Vertical cylindrical vessels under combined loading (for internal or external, or both pressures)

Calculation in accordance with this Clause (3.7.5) is not necessary for many vessels and is only required for tall vessels where additional stresses due to combined loading become significant.

The minimum calculated thickness of vertical cylindrical vessels subject to combined loading shall be calculated using the equations below, but in addition the calculated thickness shall be no less than required by Clause 3.7.3. These equations adopt the basis that the stress equivalent to the membrane stress shall nowhere exceed the design strength.

The loadings include those referred to in Clause 3.2.3 which cause bending, or axial stresses, or both, in addition to those due to internal and external pressure.

The normal operating condition might not be the most critical. The out-of-service condition, with pressure terms equal to zero, or the hydrostatic test condition including force due to standard gravity acting on the mass of water might be the governing condition. The need to allow for simultaneous application of full wind loading during hydrostatic testing shall be examined to suit local conditions (see Clause 3.2.3(e)).

The following stresses shall be calculated with regard to signs, tensile (positive), compressive (negative), internal pressure (positive), external pressure (negative):

$S_b = 4M/(\pi D_m^2 t \cos \alpha)$ (bending stress) . . . 3.7.5(1)

$S_c = PD_m/(2t \cos \alpha)$ (circumferential pressure stress) . . . 3.7.5(2)

$S_w = W/(\pi D_m t \cos \alpha)$ (weight stress) . . . 3.7.5(3)

S_E shall be taken as the greatest of—

$|S_c|, |S_c/2 + S_w + S_b|, |S_c/2 + S_w - S_b|, |S_w + S_b - S_c/2|, |S_w - S_b - S_c/2|$ . . . 3.7.5(4)

(i.e. equivalent stress or Tresca stress)

The above stresses shall be calculated for the worst case condition (of those listed below) and the location (height) on the vertical vessel where the stress is maximum. Note the place of maximum stress could be different in each case.

(a) Design conditions.

(b) Hydrostatic test conditions.

(c) Hydrostatic test conditions without pressure.

(d) Full weight, no pressure.

(e) Design pressure, column empty (where relevant).

(f) Zero pressure, vessel empty.

(g) Any other foreseeable operating condition.

The above stresses shall be limited as follows:

Where S_c, or $(S_c/2 + S_w + S_b)$ are tensile (positive) then:

$S_E, S_c, (S_c/2 + S_w + S_b) \leq \eta f$ or 1.5 ηf under hydrostatic test conditions . . . 3.7.5(5)

When $(S_w - S_b)$ is negative, its magnitude shall be limited to the lesser of f and S_{IL}

where $S_{IL} = 0.605E \dfrac{t \cos \alpha}{D_m} \times \dfrac{(2880 + D_m / t \cos \alpha)}{(3200 + 10D_m / t \cos \alpha)}$ . . . 3.7.5(6)

NOTE: This equation is a curve fit with a factor of safety of 2.0, taken from Baker E H, Kovalevsky, L & Rish F L, '*Structural Analysis of Shell*' Robert E Krieger Publishing Co. Malibar, 1972.

When S_c is compressive (negative) then:

$|S_c| \leq |S_{cL}|$

where

$S_{cL} = P_e D_m/2t \cos \alpha$ . . . 3.7.5(7)

where P_e is calculated according to Clause 3.9.3.

When both S_c and $(S_w - S_b)$ are negative then:

$$\frac{|S_c|}{|S_{cL}|} + \frac{|S_w - S_b|}{S_{IL}} \leq 1.0$$ . . . 3.7.5(8)

NOTES:

1 In the above equations, it is permissible to replace t with T minus corrosion allowance.

2 The above tensile and equivalent stress limits may be increased by a factor of 1.2 for conditions when wind and earthquake loadings are taken into consideration. Earthquake and wind loadings need not be considered to act simultaneously.

3 The above equations cannot be reduced to a convenient explicit expression for the calculation of t, and the solution must be by trial and error.

4 A solid section is defined as a solid plate and specifically excludes cross-sections such as a plate with stiffening ribs, a hollow section, or the whole cross-section of a vessel.

3.7.6 Horizontal cylindrical vessels under combined loading

The minimum calculated thickness of horizontal cylindrical vessels subjected to combined loading shall be determined in the same manner as for vertical cylindrical vessels, except that the force due to the mass shall be incorporated in the bending moment M and the symbol W shall be omitted. For local stresses at supports, see Clause 3.24.4.

3.7.7 Conical shells subject to internal pressure

The minimum calculated thickness of a conical shell subject to internal pressure shall be determined from Clause 3.10.

3.8 THICK-WALLED CYLINDRICAL AND SPHERICAL SHELLS SUBJECT TO INTERNAL PRESSURE

Thick-walled cylinders and spherical shells subject to pressure shall be in accordance with Clause 3.7.

For components of simple vessels not subject to additional external or internal loads and for which detailed fatigue analysis is not required by Appendix M, the minimum thickness shall be calculated from equations given in Clause 3.7, except that at the discretion of the designer, the following equations may be used:

For cylindrical shells—

$$P = f \log_e \left(\frac{D + 2t}{D} \right) \quad \text{. . . 3.8(1)}$$

or

$$t = 0.5D\left(e^{P/f} - 1\right) \quad \text{. . . 3.8(2)}$$

For spherical shells—

$$P = 2f \log_e \left(\frac{D + 2t}{D} \right) \quad \text{. . . 3.8(3)}$$

or

$$t = 0.5D\left(e^{P/2f} - 1\right) \quad \text{. . . 3.8(4)}$$

Vessels or vessel components designed to resist additional loads or requiring detailed fatigue analysis shall be the subject of a detailed stress investigation.

3.9 CYLINDRICAL AND SPHERICAL SHELLS SUBJECT TO EXTERNAL PRESSURE

3.9.1 General

The minimum thickness of cylindrical or spherical shells or cylindrical or spherical parts of vessels subject to external pressure shall be no less than that determined in accordance with this Clause (3.9), or the method given in ASME BPV-VIII-1. The thickness so determined shall be not less than that required by Clause 3.4.3.

This Clause applies to vessels either with or without longitudinal or circumferential joints, and either with or without stiffening rings. The possible influence of other loadings (Clause 3.2.3) shall be considered and, where necessary, the stiffness of the shell shall be suitably increased. See also Clauses 3.24 and 3.25 for supports and attachments to avoid local distortion.

The minimum calculated thickness shall be increased where necessary to meet the requirements of Clause 3.4.2.

NOTE: The equations in this Clause are designed to provide a factor of safety of approximately 2.0 against the lower bound of experimental buckling results for vessels constructed to normal manufacturing tolerances. This means that the factor of safety against theoretical elastic buckling equations is 3 for cylinders and 14 for spheres to reflect the differing sensitivities of these structures to initial imperfections.

3.9.2 Notation

For the purpose of this Clause 3.9, the following notation applies:

A_a = circumferential strain of shell or cone

A_a' = circumferential strain of stiffening ring

A_s = cross-sectional area of stiffening ring, in square millimetres

B_a' = theoretical buckling stress of stiffening ring, in megapascals

d = radial depth of stiffeners used (between flanges, if any), in millimetres

D = inside diameter of shell, in millimetres

D_m = mean diameter of shell, in millimetres

= $D_o - t$ (see Figure 3.9.2)

D_o = outside diameter of shell in the fully corroded condition, in millimetres

E = Young's modulus of shell, cone or stiffener at design temperature, in megapascals

f = design strength of shell or cone at design temperature, in megapascals

I_c = required second moment of area of the combined ring/shell on a section normal to the shell and about its neutral axis parallel to the axis of the cylindrical shell, in millimetres to the fourth power

I_r = required second moment of area of stiffening ring on a section normal to the shell and about its neutral axis parallel to the axis of the cylindrical shell, in millimetres to the fourth power.

L = effective length of cylindrical shell, in millimetres (see Figure 3.9.2)

L' = length of shell which is to be included for the calculation of the second moment of area provided by the stiffening rings, in millimetres (see Figure 3.9.6.2)

= $(D_m T)^{1/2}$ or L_s, whichever is less

L_s = sum of half distances from stiffening ring to rings on either side (for equispaced rings $L_s = L$), in millimetres

n = number of circumferential buckling lobes

P = calculation pressure (i.e. net external pressure), in megapascals (see Clause 3.2.1.3)

P_e = theoretical pressure required to cause elastic buckling of shell, in megapascals

P_y = theoretical pressure required to cause plastic yielding of shell, in megapascals

T = actual thickness (taken as nominal thickness less allowances), in millimetres.

V = radial shear load, in newtons

Q = first moment of area about the neutral axis of that part of shell which is being credited as part of the stiffener ring, in millimetres cubed

t = minimum calculated thickness of the pressure part (exclusive of added allowances, see Clause 3.4.2), in millimetres

t_f = thickness of stiffener flange, in millimetres

t_w = thickness of stiffener web, in millimetres

Y = specified minimum yield strength (0.2% proof stress) at design temperature, in megapascals. If values are not available, Y may be taken as—

$1.5f$ for carbon, low alloy and ferritic steels

$1.1f$ for austenitic steels and non-ferrous metals

$Z = \dfrac{\pi D}{2L}$

α = half apex angle of conical end or reducer, in degrees

λ = buckling wave length, in millimetres

w = outstanding width of stiffener flange from centre of web, in millimetres

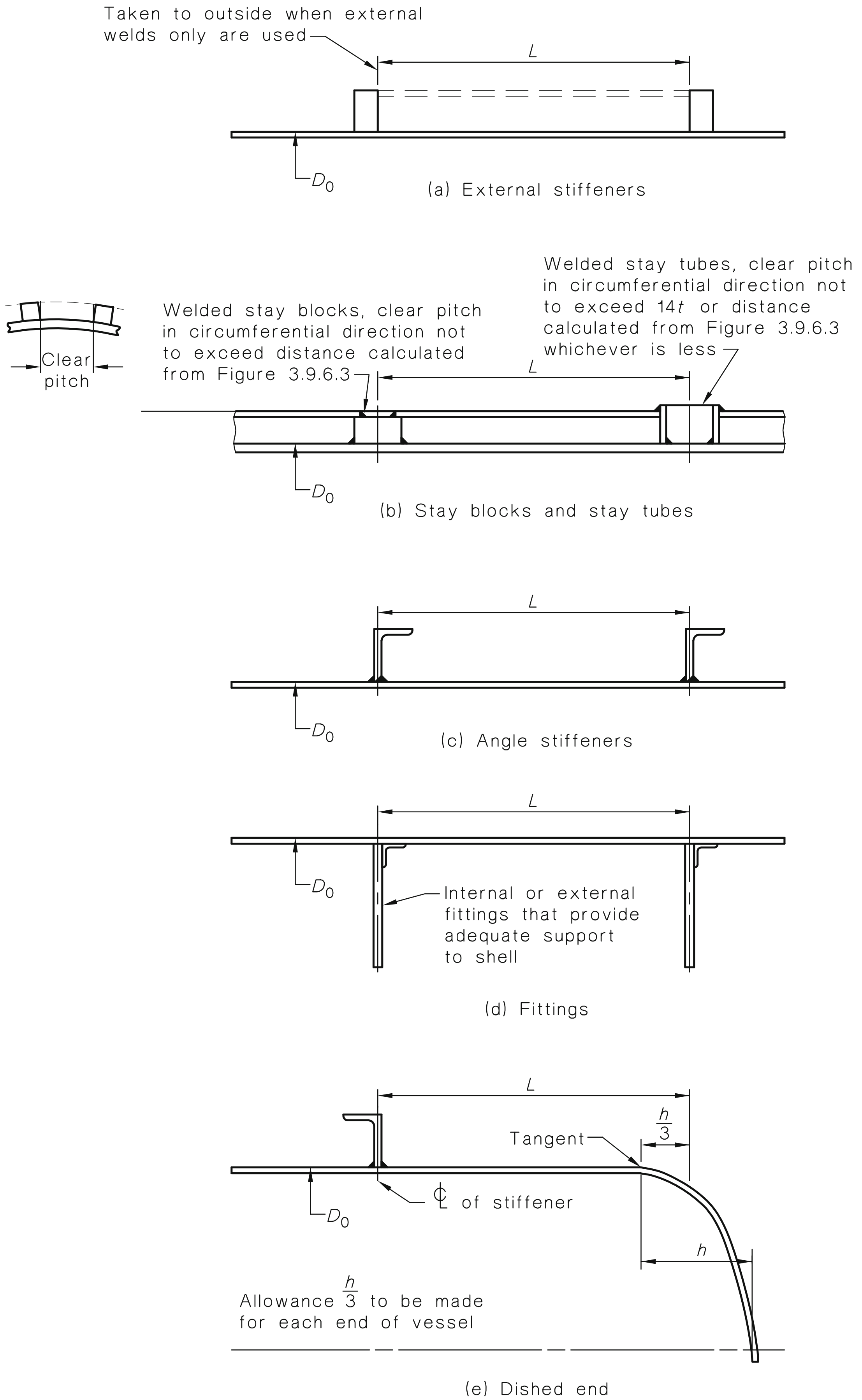
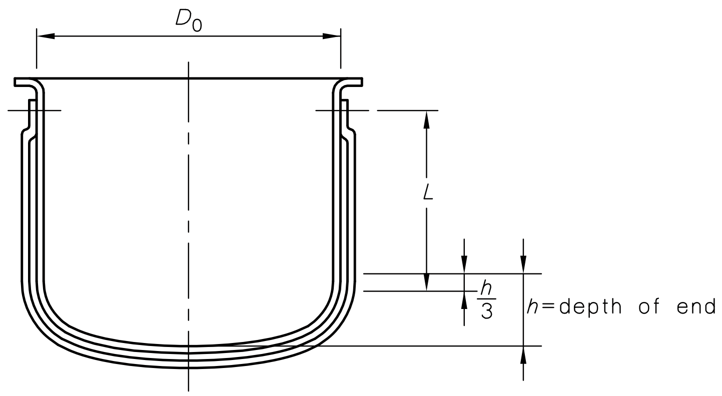
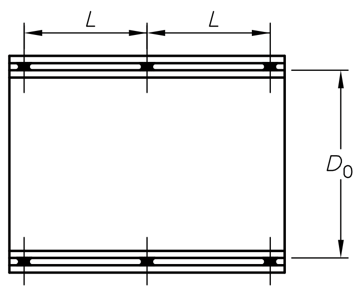


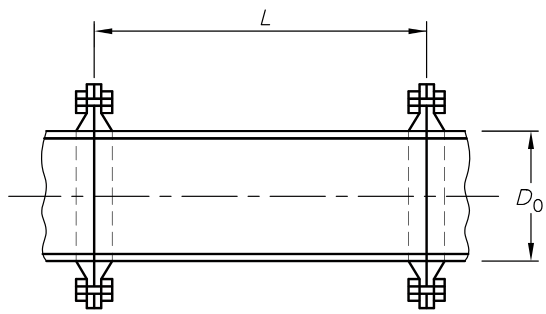
FIGURE 3.9.2 (in part) EFFECTIVE LENGTH (L) OF VESSELS SUBJECT TO EXTERNAL PRESSURE



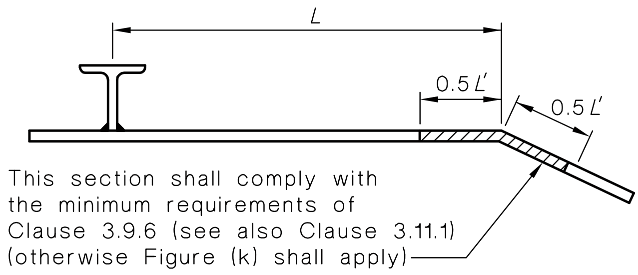
(f) Jacketed vessel



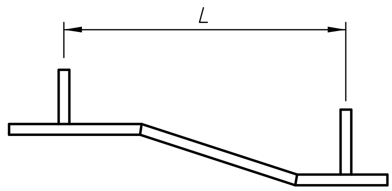
(g) Jacketed cylinder with supports



(h) Flanges



(j) Integral cone to cylinder stiffening



NOTES:

1 When the cone-to-cylinder or the knuckle-to-cylinder junction is not a line of support, the nominal thickness of the cone, knuckle, or toriconical section shall not be less than the minimum required thickness of the adjacent cylindrical shell.

2 Calculations shall be made using the diameter and corresponding thickness of each section with dimension L as shown.

(k) Conical section or end

FIGURE 3.9.2 (in part) EFFECTIVE LENGTH (L) OF VESSELS SUBJECT TO EXTERNAL PRESSURE

3.9.3 Cylindrical shells

The minimum calculated thickness of a cylindrical shell, either seamless or with butt joints, under external pressure shall not be less than that determined from the following procedure:

(a) Assume a value for t and determine a value of A_a as follows:

(i) A conservative value of A_a may be taken as the greater of the values given by the following equations:

$$A_a = \frac{1.3t^{1.5}}{D_m^{0.5}L} \quad \ldots 3.9.3(1)$$

$$A_a = 1.1\left(\frac{t}{D_m}\right)^2 \quad \ldots 3.9.3(2)$$

(ii) Where greater accuracy is required, A_a may be calculated from Equation 3.9.3(3).

$$A_a = \frac{1}{n^2 - 1 + \frac{Z^2}{2}}\left[\left(\frac{Z^2}{n^2 + Z^2}\right)^2 + \frac{t^2\left(n^2 - 1 + Z^2\right)^2}{2.73D_m^{\ 2}}\right] \quad \ldots 3.9.3(3)$$

where

n = number of circumferential buckling lobes and is an integer ≥2 which minimizes the value of A_a, determined by iterative application of Equation 3.9.3(3).

A first approximation of the value of n may be determined from Equation 3.9.3(4) but shall not be less than 2.

$$n = Z\left[\frac{L}{(D_m t)^{0.5} - 1}\right]^{0.5} \quad \ldots 3.9.3(4)$$

NOTE: This value of n is used in determining the stiffening parameters.

(b) Determine values of P_e and P_y from Equations 3.9.3(5) and 3.9.3(6), respectively.

$$P_e = \frac{2EA_a t}{D_m} \quad \ldots 3.9.3(5)$$

$$P_y = \frac{2Yt}{D_m} \quad \ldots 3.9.3(6)$$

(c) Calculate the value of the maximum permissible calculation pressure, P, for the value of t assumed in Item (a) from Equation 3.9.3(7) or Equation 3.9.3(8), as applicable.

where

$P_e \leq P_y$

$$P = \frac{P_e}{3} \quad \ldots 3.9.3(7)$$

where

$P_e > P_y$

$$P = \frac{P_y\left(2 - P_y / P_e\right)}{3} \quad \ldots 3.9.3(8)$$

(d) If the value of P so obtained is less than the required calculation pressure, the assumed value of t shall be increased and the procedure repeated until the value of P obtained is equal to or greater than the required calculation pressure.

NOTE: The equations here are designed to provide a factor of safety of approximately 2.0 against the lower bound of experimental buckling results for vessels constructed to normal manufacturing tolerances. This means the factor of safety against theoretical buckling equations is 3 for cylinders and 14 for spheres to reflect the differing sensitivities of these structures to initial imperfections.

3.9.4 Spherical shells

The minimum calculated thickness of spherical shells under external pressure, either seamless or with butt joints, shall not be less than that determined from the following procedure:

(a) Assume a value for t and calculate values for P_e and P_y from Equations 3.9.4(1) and 3.9.4(2), respectively.

$$P_e = 4.84E\left(\frac{t}{D_m}\right)^2 \quad \text{. . . 3.9.4(1)}$$

$$P_y = 4\frac{Yt}{D_m} \quad \text{. . . 3.9.4(2)}$$

(b) Calculate the value of the maximum permissible external pressure for the value of t assumed in Item (a) from Equation 3.9.4(3) or Equation 3.9.4(4), as applicable.

where

$P_e \leq P_y$

$$P = 0.07P_e \quad \text{. . . 3.9.4(3)}$$

where

$P_e > P_y$

$$P = 0.07P_y\left(5 - \frac{16}{3 + P_e / P_y}\right) \quad \text{. . . 3.9.4(4)}$$

(c) If the value of P so obtained is less than the required calculation pressure, the assumed value of t shall be increased and the procedure repeated until the value of P obtained is equal to or greater than the required calculation pressure.

3.9.5 Shells subject to external pressure and combined loadings

Cylindrical shells subject to external pressure and combined loadings, in addition to satisfying the requirements of this Clause (3.9), shall also satisfy Clause 3.7.5 (vertical vessels) or Clause 3.7.6 (horizontal vessels). In both the latter Clauses the sign of P shall be negative.

Where necessary, vessels shall be provided with stiffeners or other additional means of support to prevent overstress or excessive distortion due to external loadings listed in Clause 3.2.3.

3.9.6 Stiffening rings for cylindrical shells subject to external pressure

3.9.6.1 *Second moment of area*

Stiffening rings consisting of internal or external diaphragms or structural sections may be used to limit the effective length of a cylindrical shell subject to external pressure. The required second moment of area and the available second moment of area of the stiffener shall be determined in accordance with Items (a) and (b), respectively, and shall comply with Item (c) as follows:

(a) The required second moment of area of a circumferential stiffening ring shall be not less than that determined from Equation 3.9.6(4) or Equation 3.9.6(5) as applicable, in accordance with the following procedure:

$$B_a' = \frac{1.5PD_m}{t + A_s / L_s} \quad \ldots 3.9.6(1)$$

$$\text{For } B_a' < Y,\ A_a' = \frac{B_a'}{E} \quad \ldots 3.9.6(2)$$

$$\text{For } B_a' \geq Y,\ A_a' = \frac{Y}{E\left(2 - B_a' / Y\right)} \quad \ldots 3.9.6(3)$$

$$I_r \geq \frac{D_m^{\ 2} L_s A_a' \left(t + A_s / L_s\right)}{14} \quad \ldots 3.9.6(4)$$

$$I_c \geq \frac{D_m^{\ 2} L_s A_a' \left(t + A_s / L_s\right)}{10.9} \quad \ldots 3.9.6(5)$$

Where the stiffening ring is not attached to the shell, or where the stiffening ring is attached but only the ring is considered, I_r determined from Equation 3.9.6(4) is applicable.

Where the stiffening ring is attached to the shell and part of the shell is credited in the actual second moment of area of the combined shell/ring, I_c determined from Equation 3.9.6(5) is applicable.

(b) The available second moment of area of a circumferential stiffening ring shall be calculated using the same cross-sectional area as that used to determine I_r or I_c as applicable.

Where I_c is the applicable required second moment of area, the length L' of the shell plate (taken as one half on each side of the ring centroid) may be included as part of the cross-section of the stiffener provided that such length contributes area to only one ring and the stiffening ring is effectively welded to the shell.

(c) If the required second moment of area determined from Item (a) is greater than the available second moment of area calculated from Item (b), a new size of stiffener with a larger second moment of area shall be selected and the procedures of Items (a) and (b) repeated.

3.9.6.2 *Form of stiffening rings*

Stiffening rings shall extend completely around the circumference of the shell except as provided in Clause 3.9.6.3.

Each joint between the ends or sections of rings shall be made so the required second moment of area of the ring is maintained. (See Figure 3.9.6.2.)

Internal plane structures perpendicular to the longitudinal axis of the cylinder, such as bubble trays, baffle plates or diaphragms, may be considered to act as stiffening rings provided they are suitably designed for both purposes. Internal diaphragms used as stiffening rings and subject to transverse pressure shall be designed to support the loads due to the pressure on the diaphragm and on the effective length of the shell, consideration being given to the buckling of the diaphragm under the edge load using a factor of safety of three against buckling and making allowance for the attached or free edge condition.

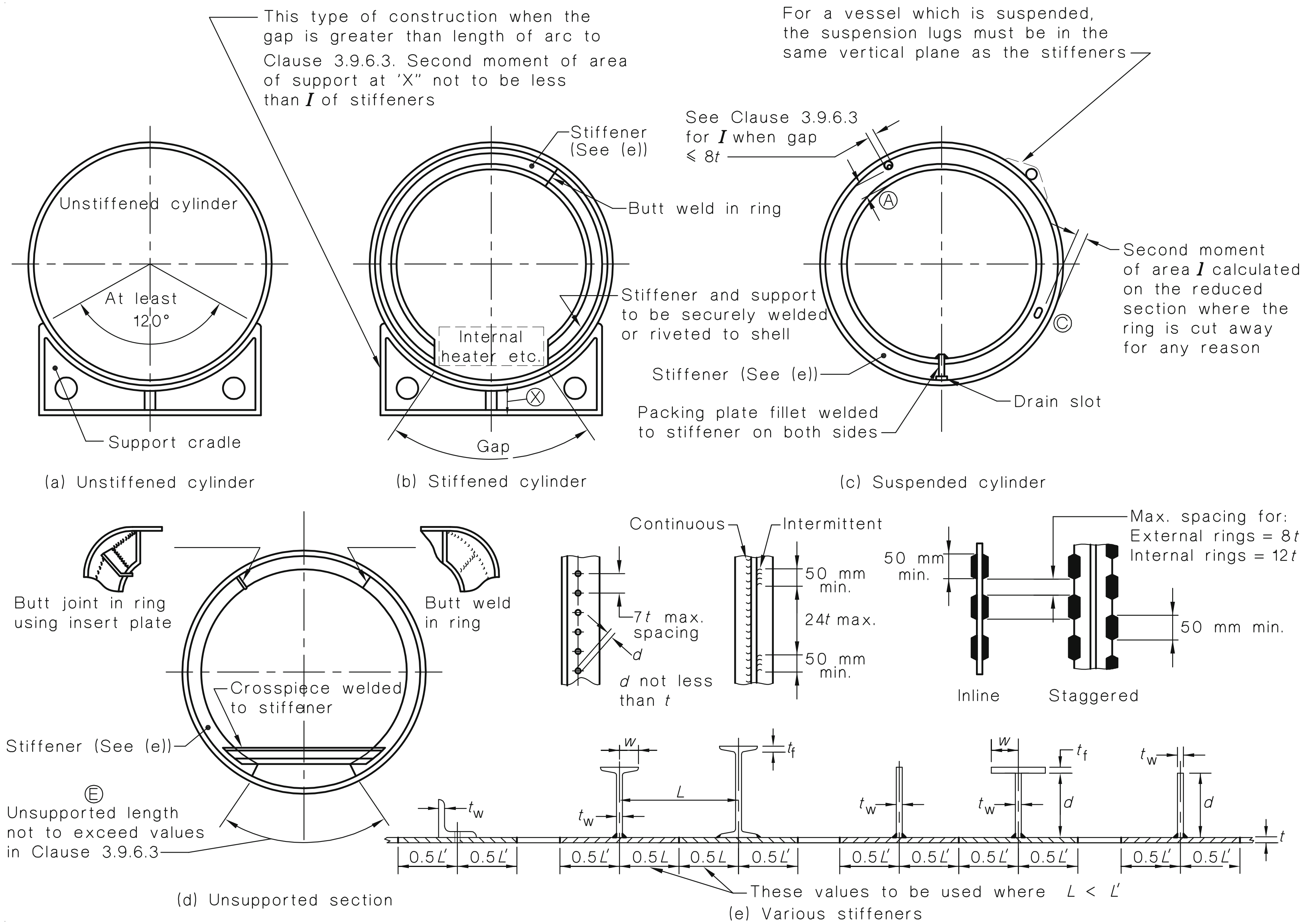


FIGURE 3.9.6.2 STIFFENING RINGS FOR CYLINDRICAL VESSELS SUBJECT TO EXTERNAL PRESSURE

To ensure lateral stability, stiffening rings (whether on the inside or outside of the vessel) shall comply with the following limiting proportions (see Note):

(a) For stiffeners flanged at the edge remote from the shell surface—

$$\frac{d}{t_w} \leq 1.1\sqrt{\frac{E}{R_{e(T)}}} \text{ ; and} \quad \ldots 3.9.6(6)$$

$$\frac{w}{t_f} \leq 0.5\sqrt{\frac{E}{R_{e(T)}}} \quad \ldots 3.9.6(7)$$

(b) For flat bar stiffener—

$$\frac{d}{t_w} \leq 0.5\sqrt{\frac{E}{R_{e(T)}}} \quad \ldots 3.9.6(8)$$

NOTE: The values of d and w, so determined, are the maxima that can be used for determining the required I_r and I_c. The actual dimensions used in manufacture should not greatly exceed these values.

Where the effective length of a shell is determined by a row of screwed or welded stays or stay blocks, the stay diameter shall not be less than twice the thickness of the shell plate, and the maximum unsupported arc of shell, measured between stay centres, shall comply with Clause 3.9.6.3.

Corrosion-resistant linings shall not be included in the calculation for the wall thickness except where permitted by Clause 3.3.1.2.

3.9.6.3 *Local spaces in stiffening rings*

Stiffening rings having local spaces between the ring and the shell (as shown at A and E in Figure 3.9.6.2) shall not have any unsupported arc of the shell exceeding the length of arc specified below unless additional reinforcement is provided as shown at X in Figure 3.9.6.2 or unless—

(a) the unsupported shell arc does not exceed 90°; and

(b) the unsupported shell arcs in adjacent stiffening rings are staggered 180°; and

(c) the dimension L defined in Figure 3.9.2 is taken as larger of—

  (i) the greatest distance between alternate stiffening rings; and

  (ii) the distance from the end tangent line to the second stiffening ring plus 0.33 times the end depth.

The maximum unsupported arc length shall not exceed $\frac{\lambda}{4}$

where

$$\lambda = \frac{\pi D_m}{n} \quad \ldots 3.9.6(9)$$

n = number of circumferential buckling lobes and is an integer ≥2 that minimizes the value of A_a (see Clause 3.9.3(a)(ii)).

Stiffening rings with holes or spaces in the ring as shown at A and C in Figure 3.9.6.2, shall be suitably reinforced so that the second moment of area required for the ring in A or the combined ring/shell section in C is maintained within the sections indicated. The second moment of area of each section shall be taken about its own neutral axis. Where the gap at A does not exceed eight times the thickness of the shell, the combined second moment of area of the ring and shell section may be used.

3.9.6.4 *Attachment of stiffening rings*

Stiffening shall be attached as follows:

(a) Stiffening rings, when used, may be placed on the inside or outside of a vessel. Internal rings need not be attached to the shell provided they have adequate lateral support. The attachment of rings to the shell shall be by welding, brazing, riveting or bolting. Brazing may be used if the vessel is not to be subsequently stress-relieved. The ring shall be in contact with the shell around the circumference.

(b) Stiffening rings may be attached to the shell by either continuous or intermittent welding. The total length of the intermittent welding on each side of the stiffening ring shall be—

    (i) no less than one-half of the outside circumference of the vessel for rings on the outside; and

    (ii) no less than one-third of the circumference of the vessel for rings on the inside. Acceptable arrangements and spacings of intermittent welds are shown in Figure 3.9.6.2.

(c) Where the rings are on the outside and riveted to the shell—

    (i) the nominal diameter of the rivets shall be no less than the thickness of the shell plate; and

    (ii) the centre-to-centre distance of the rivet holes shall not exceed that required by Figure 3.9.6.2.

(d) Where the stiffening ring and shell are subject to corrosion, the stiffening ring shall be attached to the shell by continuous welds on both sides.

3.9.6.5 *Strength of attachment welds*

Welds attaching a stiffener ring shall be sized to resist the combination of:

(a) Full radial pressure load from the shell between stiffeners. This equals PL_s, in newtons/millimetre.

(b) Shear loads acting radially across the stiffener from any external design load, in newtons/millimetre.

(c) Radial shear load, V, equal to 2% of the ring's compressive load, i.e. 0.01 $PL_s\,D_o$, in newtons. This value results in a weld load equal to $\frac{VQ}{I_c}$, in newtons/millimetre.

The combined weld load = $\left[(PL_s)^2 + \left(\frac{VQ}{I_c}\right)^2\right]^{0.5}$ in newtons/millimetre.

The fillet welds shall be sized so that—

(i) the total throat area is sufficient to withstand the combined weld load without exceeding the allowable shear stress; and

(ii) the minimum leg length is not less than the smallest of 6 mm, the vessel thickness at the stiffener, or the stiffener thickness.

3.10 CONICAL ENDS AND REDUCERS SUBJECT TO INTERNAL PRESSURE

3.10.1 General

Conical ends and reducers subject to internal pressure shall be designed in accordance with this Clause (3.10). The minimum calculated thickness shall be increased where necessary to meet the requirements of Clauses 3.4.2 and 3.4.3 and to meet other appropriate loadings given in Clause 3.2.3.

Conical ends and reducers may be constructed in several ring sections of decreasing thickness as determined by the corresponding decreasing diameter.

This Clause applies to conical ends or reducers that are concentric with the cylindrical shell and where all the longitudinal loads are transmitted wholly through the conical section.

NOTE: It may be assumed that this Clause applies also to an offset cone such as a reboiler provided all parts of the cone fall within the projected perimeter of the large end.

3.10.2 Notation

For purposes of Clauses 3.10.3 to 3.10.5, the following notation applies:

D_1 = inside diameter of conical section or end at the position under consideration, i.e. D_1 may vary between D_S and D_L (see Figure 3.10.2), in millimetres

D_{mL} = mean diameter of conical end or reducer at the large end, in millimetres

= $D_L + t$ (see Figure 3.10.2)

D_{mS} = mean diameter of conical end or reducer at the small end, in millimetres

f = design tensile strength at the calculation temperature (see Table B1), in megapascals

P = calculation pressure (see Clause 3.2.1), in megapascals

r_L = inside radius of knuckle at larger cylinder, in millimetres

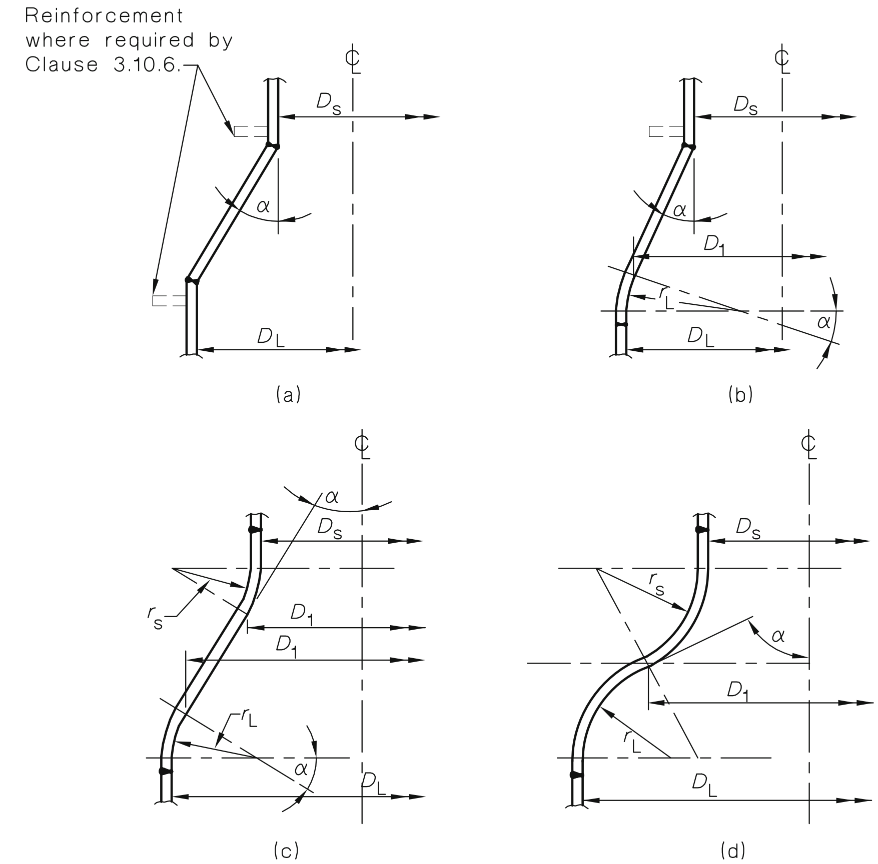
r_S = inside radius of knuckle at smaller cylinder, in millimetres

t = minimum calculated thickness of conical ends or reducers (exclusive of added allowances—see Clause 3.4.2), in millimetres

α = angle of slope of conical end or reducer (at the point under consideration) to the vessel axis (see Figure 3.10.2), in degrees

NOTE: For offset cones, use the larger α.

η = lowest efficiency of any joint in the conical ends or reducers (see Clause 3.10.4 for attachment joints)



$r_L \geqslant$ greater of $(0.06(D_L+2t))$ and $(3t)$

FIGURE 3.10.2 CONICAL ENDS AND REDUCERS

3.10.3 Conical sections

The minimum calculated thickness of a conical section shall be determined by—

$$t = \frac{PD_1}{2f\eta - P} \times \frac{1}{\cos\alpha} \qquad \text{. . . 3.10.3(1)}$$

or

$$P = \frac{2\, f\eta t \cos\alpha}{D_1 + (t\cos\alpha)} \qquad \text{. . . 3.10.3(2)}$$

When the angle, α, exceeds 70°, the thickness of the conical section shall be determined as for a flat end as specified in Clause 3.15.

3.10.4 Attachment of cone to cylinder

A transition knuckle is recommended between the cone and the cylinder. A transition knuckle shall be used when the angle, α, is greater than 30°, except where additional analysis to Clause 3.1.3(c) shows it is not required. See Clause 3.10.5 for minimum calculated thickness.

Where the angle α does not exceed 30° the cone may be attached to the cylinder without a transition knuckle provided the attachment is a double butt-welded joint and complies with the requirements of Clause 3.10.6.

Alternative cone/cylinder junctions are permissible (for example a ring beam reinforcement in the absence of a transition knuckle) provided the appropriate limits on stress intensity (see Figure H1) are complied with, and a safety factor of not less than 2 is achieved on buckling collapse based on the corroded thickness of the components.

3.10.5 Transition knuckles

The minimum calculated thickness of a transition knuckle between the large end of the cone and the cylinder shall be at least equal to the thickness required for a torispherical end as determined in Clause 3.12.5.2, substituting—

$$\frac{D_1}{2\cos\alpha} \text{ for } R \qquad \ldots 3.10.5$$

The transition knuckle shall have a length of straight flange sufficient to meet the requirements shown in Figure 3.12.6.

Any taper between the knuckle and cone shall be in accordance with Figure 3.5.1.8.

Transition knuckles at the small end of the cone shall have a minimum actual thickness at least equal to the minimum required thickness of the cylinder to which they attach.

'Reverse curve' transition knuckles shown in Figure 3.10.2(d) may be used provided their design conforms to the requirements of Clause 1.5.

Conical sections in Group F or Group G steels shall have transition knuckles at both ends, terminating in skirts (or extensions). The knuckle radius shall be no less than 10% of the outside diameter of the skirt, or no less than three times the cone thickness, whichever is larger. The skirt length shall be not less than $0.50\sqrt{rt}$ (where r is the inside radius of the adjacent cylinder and t is the thickness of the cone) or not less than 38 mm, whichever is larger.

3.10.6 Reinforcement

3.10.6.1 *General*

Reinforcement may be required where the cone attaches to the cylinder without a transition knuckle, as provided in Clause 3.10.4. Where reinforcement is required, it shall be in accordance with this Clause (3.10.6).

3.10.6.2 *Notation*

For the purposes of Clause 3.10.6 the following notation applies:

t = minimum calculated thickness of cylinder at cone-to-cylinder junction, (exclusive of added allowance, see Clause 3.4.2), in millimetres.

T_s = nominal thickness of cylinder at cone-to-cylinder junction, exclusive of corrosion allowance, in millimetres

T_c = nominal thickness of cone at cone-to-cylinder junction, exclusive of corrosion allowance, in millimetres

T_e = the smaller of $(T_s - t)$ and $[T_c - (t/\cos\alpha)]$, in millimetres

D_S = inside diameter of small cylinder, in millimetres

D_L = inside diameter of large cylinder, in millimetres

A = required area of reinforcement, in square millimetres

A_e = effective area of reinforcement due to excess metal thickness, in square millimetres

Δ = value to indicate need for reinforcement at cone-to-cylinder intersection having an angle α ≤30°; when Δ ≥ α, no reinforcement at the junction is required (see Tables 3.10.6.3 and 3.10.6.4)

$$m = \text{the smaller of } \left(\frac{T_S}{t}\cos\,(\alpha - \Delta)\right) \text{ and } \left(\frac{T_c \cos\,\alpha \cos\,(\alpha - \Delta)}{t}\right) \qquad \text{. . . 3.10.6.2}$$

η = lowest efficiency of the longitudinal joint in the shell or end or of the joint in the reinforcement ring; for the large end of the reducer in compression η = 1 for butt welds.

P, f and α are as defined in Clause 3.10.2.

3.10.6.3 *Reinforcement at large end of cone to cylinder*

Reinforcement shall be provided at the junction of the cone with the large cylinder of conical ends and reducers without knuckles when the value of Δ obtained from Table 3.10.6.3 using the appropriate ratio *P/f*η is less than α. Intermediate values may be interpolated.

TABLE 3.10.6.3

VALUES OF Δ FOR JUNCTION AT THE LARGE CYLINDER FOR α ≤ 30°

*P/f*η	0.001	0.002	0.003	0.004	0.005
Δ, degrees	11	15	18	21	23
*P/f*η	0.006	0.007	0.008	0.009*	
Δ, degrees	25	27	28.5	30	

* Δ = 30° for greater values of *P/f*η.

The cross-sectional area of the reinforcement ring shall be at least equal to that determined by Equation 3.10.6.3(1), as follows:

$$A = \frac{PD_L^2}{8f\eta}\left(1 - \frac{\Delta}{\alpha}\right)\tan\alpha \qquad \text{. . . 3.10.6.3(1)}$$

When the thickness, less corrosion allowance, of both the reducer and cylinder exceeds that required by the applicable design equation, the minimum excess thickness may be considered to contribute to the required reinforcement ring in accordance with the following equation:

$$A_e = 2.8\,T_e\sqrt{(D_L\,T_S)} \qquad \text{. . . 3.10.6.3(2)}$$

The additional area of reinforcement as required shall be situated within a distance of $0.7\sqrt{(D_L T_S)}$ from the junction of the reducer and the cylinder. The centroid of the added area shall be within a distance of $0.35\sqrt{(D_L T_S)}$ from the junction.

3.10.6.4 *Reinforcement at small end of cone to cylinder*

Reinforcement shall be provided at the junction of the cone with the small cylinder of conical ends and reducers without knuckles when the value of the Δ obtained from Table 3.10.6.4 using the appropriate ratio *P/f*η, is less than α. Intermediate values may be interpolated.

TABLE 3.10.6.4

VALUES OF Δ FOR JUNCTIONS AT THE SMALL CYLINDER FOR α ≤30°

$P/f\eta$	0.002	0.005	0.010	0.02
Δ, degrees	4	6	9	12.5
$P/f\eta$	0.04	0.08	0.10	0.125*
Δ, degrees	17.5	24	27	30

* Δ = 30° for greater values of $P/f\eta$.

The cross-sectional area of the reinforcement ring shall be at least equal to that determined by the following equation:

$$A = \frac{PD_S^2}{8f\eta}\left(1-\frac{\Delta}{\alpha}\right)\tan\alpha \qquad \ldots 3.10.6.4(1)$$

When the thickness, less corrosion allowance, of either the reducer or cylinder exceeds that required by the applicable design equation, the excess thickness may be considered to contribute to the required reinforcement ring in accordance with the following equation:

$$A_e = 0.7m(D_S t)^{1/2}\left[\left(T_c - \frac{t}{\cos\alpha}\right) + (T_S - t)\right] \qquad \ldots 3.10.6.4(2)$$

The additional area of reinforcement as required shall be situated within a distance of $0.7(D_S T_C)^{1/2}$ from the junction and the centroid of the added area shall be within a distance of $0.35(D_S T_C)^{1/2}$ from the junction.

3.11 CONICAL ENDS AND REDUCERS SUBJECT TO EXTERNAL PRESSURE

3.11.1 General

The minimum calculated thickness of conical ends and reducers subject to external pressure, i.e. on the convex side, shall be no less than that required in Clause 3.11.2, and no less than that required by Clause 3.10 for an internal pressure equal to the external pressure, assuming η = 1.0. The minimum calculated thickness shall be increased where necessary to meet the requirements of Clauses 3.4.2 and 3.4.3 and to meet other appropriate loadings given in Clause 3.2.3.

3.11.2 Minimum calculated thickness

The minimum calculated thickness of a conical end or reducer under external pressure, either seamless or with butt joints, may be determined from Clause 3.9.3 by calculating P_e and P_y for cylinders of the following equivalent dimensions:

(a) Determine the value of P_e using—

(i) Equivalent mean diameter D_m, of cylinder = $\frac{(D_{mL}+D_{mS})}{2}$ . . . 3.11.2(1)

(ii) Equivalent thickness, $t_e = T\cos\alpha$ . . . 3.11.2(2)

(iii) Equivalent length, L_e = the axial length between centres of stiffeners or equivalent. (See Figures 3.9.2(j) and (k), dimension L).

(b) Determine the value of P_y using—

(i) Equivalent mean diameter, $D_m = D_{mL}$ . . . 3.11.2(3)

(ii) Equivalent thickness, $t_e = T\cos\alpha$ . . . 3.11.2(4)

To avoid yielding, the junction of the cone shall also comply with the requirements of Clause 3.10 for the design external pressure.

The nomenclature is the same as that given in Clauses 3.9.2 and 3.10.2.

3.12 DISHED ENDS SUBJECT TO INTERNAL PRESSURE

3.12.1 General

Unstayed dished ends of spherical, ellipsoidal, or torispherical shape, subject to internal pressure (i.e. pressure on the concave side), shall be designed in accordance with this Clause (3.12). Ends constructed of Group F or Group G steels shall be spherical or ellipsoidal in shape.

3.12.2 Notation

For the purpose of Clauses 3.12 and 3.13, the following notation applies:

t = minimum calculated thickness of end at thinnest point after forming (exclusive of added allowances, see Clause 3.4.1), in millimetres

P = calculation pressures (see Clause 3.2.1), in megapascals

D = inside diameter of end in millimetres

D_o = outside diameter of end in millimetres

R = inside spherical or crown radius of end, in millimetres

R_o = outside spherical or crown radius of end, in millimetres

r = inside knuckle radius, in millimetres

η = lowest efficiency of any joint in an end including the end to shell joint in the case of an end not having a straight flange

= 1.0 for end made from one plate having a straight flange

f = design tensile strength at the design temperature (see Table B1), in megapascals

h = one-half of the length of the inside minor axis of an ellipsoidal end, or the inside depth of a torispherical end measured from the tangent line in the fully corroded condition, in millimetres

h_o = one-half of the length of the outside minor axis of an ellipsoidal end measured from the tangent line, in millimetres

K = a factor in the equation for ellipsoidal ends, depending on the end proportion $D/2h$

$= \frac{1}{6}\left[2+\left(\frac{D}{2h}\right)^2\right]$ (see Table 3.12.5.1.)

M = a factor in the equation for torispherical ends depending on the end proportion R/r

$= \frac{1}{4}\left[3+\left(\frac{R}{r}\right)^{1/2}\right]$ (see Table 3.12.5.2.)

3.12.3 Shape limitations

The shape of typical ends is shown in Figure 3.12.3.

Dished ends with reverse knuckles may be used provided the calculation pressure for the end is determined in accordance with Clause 5.12.

The inside crown radius to which an unstayed end is dished shall be no greater than the outside diameter of the end at the tangent line.

The possibility of buckling due to the setting up of high localized stresses during hydrostatic testing shall be considered when the following limits are approached or exceeded:

(a) *For ellipsoidal ends*: $\frac{D}{t} \geq 600$

(b) *For torispherical ends* with knuckle radius approaching the minimum permitted (6% of crown radius):

$$\frac{D}{t} > 100 \text{ or } P \geq 690 \text{ kPa}$$

Where an end is formed with a flattened spot or surface, the diameter of the flat spot shall not exceed that permitted for unstayed flat ends in Clause 3.15, using $K = 5$.

NOTE: For torispherical ends with $\frac{D}{t_k} > 300$, it is recommended that—

$$\frac{P}{f} \leq \frac{150\left(\frac{r}{D}\right)^{0.84}}{\left(\frac{D}{t_k}\right)^{1.53}\left(\frac{R}{D}\right)^{1.1}}$$

where

t_k = minimum calculated thickness of knuckle (exclusive of added allowances, see Clause 3.4.2), in millimetres.

For other notation, see Clause 3.12.2.

This equation applies below the creep range.

Dished ends may be one-piece or multi-piece. See Clause 3.18.7.2 for reduced thickness of the central 80% of the diameter of the end.

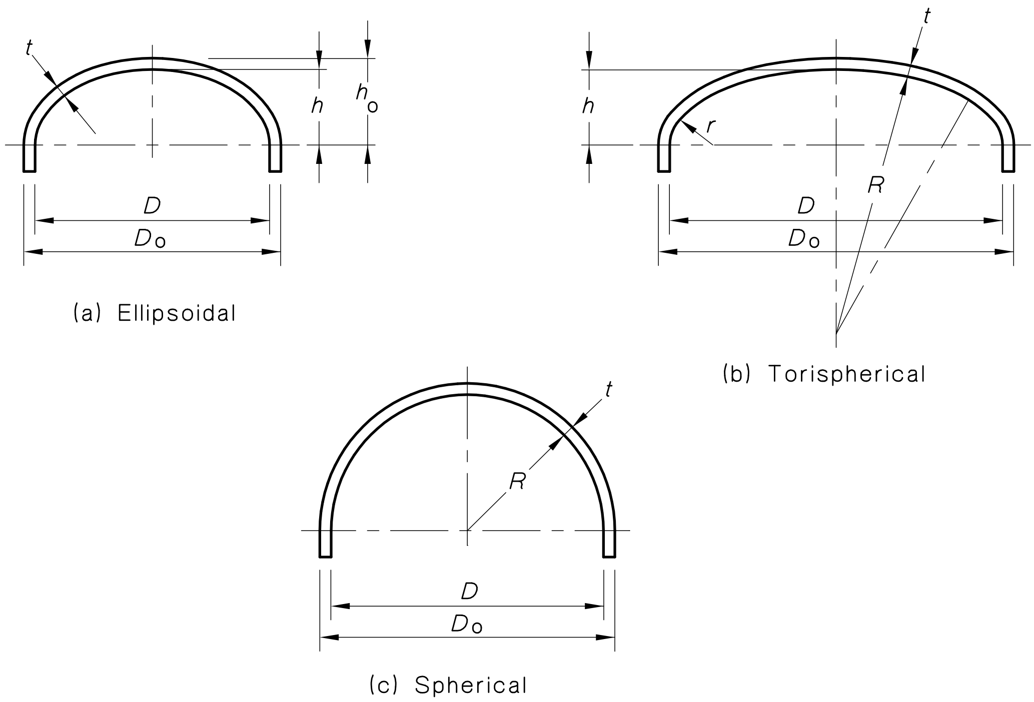


FIGURE 3.12.3 DIMENSIONS OF DISHED ENDS

3.12.4 Openings in ends

Openings in ends shall comply with the requirements of Clause 3.18.

3.12.5 Thickness of ends

3.12.5.1 *Ellipsoidal ends*

The minimum calculated thickness of ellipsoidal ends, with or without openings, shall be determined by the following equation:

$$t = \frac{PDK}{2f\eta - P} \qquad \ldots 3.12.5.1$$

where K is found from Table 3.12.5.1.

3.12.5.2 *Torispherical ends*

The minimum calculated thickness of torispherical ends, with or without openings, shall be determined by the following equation:

$$t = \frac{PRM}{2f\eta - 0.5P} \qquad \ldots 3.12.5.2$$

where M is found from Table 3.12.5.2.

TABLE 3.12.5.1

VALUES OF FACTOR *K*

(Use nearest value of *D/2h*; interpolation unnecessary)

$\frac{D}{2h}$	3.0	2.9	2.8	2.7	2.6	2.5	2.4	2.3	2.2	2.1	2.0*
K	1.83	1.73	1.64	1.55	1.46	1.37	1.29	1.21	1.14	1.07	1.00
$\frac{D}{2h}$	1.9	1.8	1.7	1.6	1.5	1.4	1.3	1.2	1.1	1.0	
K	0.93	0.87	0.81	0.76	0.71	0.66	0.61	0.57	0.50	0.50	

* Usually referred to as a 2:1 ellipsoidal end.

TABLE 3.12.5.2

VALUES OF FACTOR *M*

(Use nearest value of *R/r*; interpolation unnecessary)

$\frac{R}{r}$	1.0	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00
M	1.00	1.03	1.06	1.08	1.10	1.13	1.15	1.17	1.18
$\frac{R}{r}$	3.25	3.50	4.0	4.5	5.0	5.5	6.0	6.5	—
M	1.20	1.22	1.25	1.28	1.31	1.34	1.36	1.39	—
$\frac{R}{r}$	7.0	7.5	8.0	8.5	9.0	9.5	10.0	10.5	—
M	1.41	1.44	1.46	1.48	1.50	1.52	1.54	1.56	—
$\frac{R}{r}$	11.0	11.5	12.0	13.0	14.0	15.0	16.0	16.66*	—
M	1.58	1.60	1.62	1.65	1.69	1.72	1.75	1.77	—

* Maximum ratio allowed when R equals the outside diameter (D_o) of the end.

3.12.5.3 *Spherical ends*

The minimum thickness of spherical ends, with or without openings, shall be determined by the following equation:

$$t = \frac{PR}{2f\eta - 0.5P} \qquad \text{. . . 3.12.5.3}$$

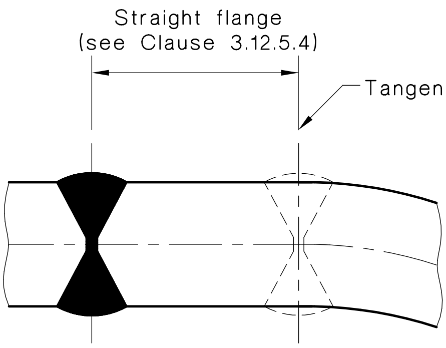
3.12.5.4 *Straight flange on ends*

The minimum calculated thickness of any cylindrical portion of an end shall comply with the appropriate requirements for cylindrical shells, including any applicable joint efficiency.

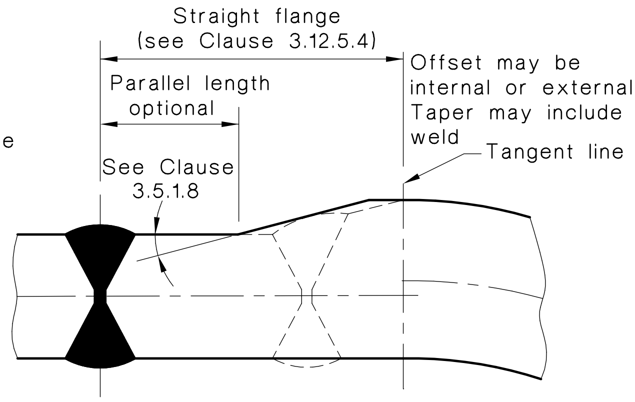
3.12.6 Attachment of ends

Ends intended for attachment by welding shall comply with Figure 3.12.6 and for Group F or Group G steels shall be attached by full penetration welds in accordance with Figure 3.12.6(a), (b), (c), (d) or (e).

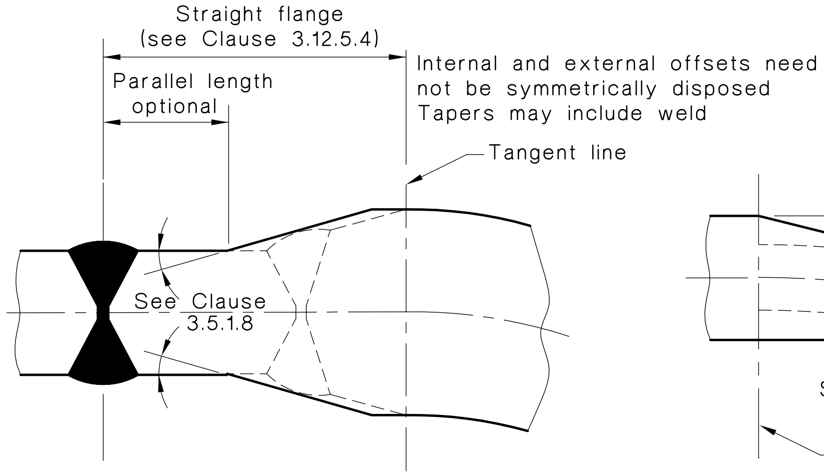
Ends intended for attachment by brazing shall have a straight flange sufficient to meet the requirements for circumferential joints in Clause 3.5.



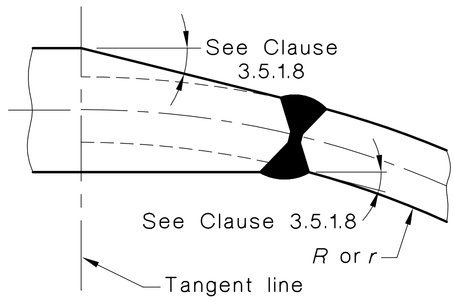
(a) Butt joint—end and shell thickness equal



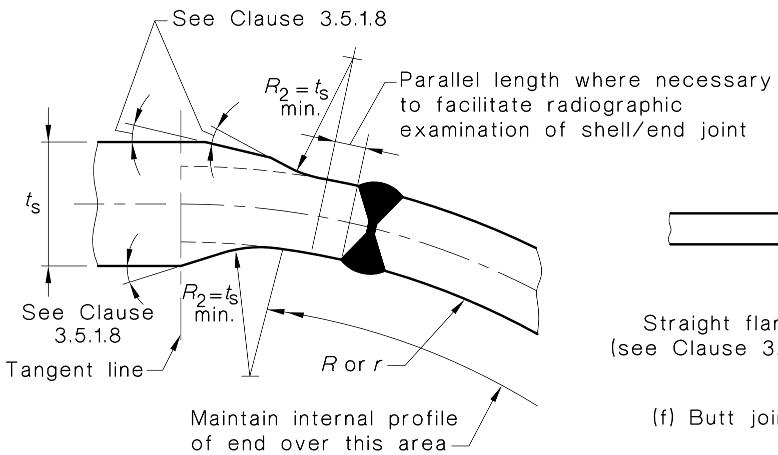
(b) Butt joint—end thicker than shell; median plane offset



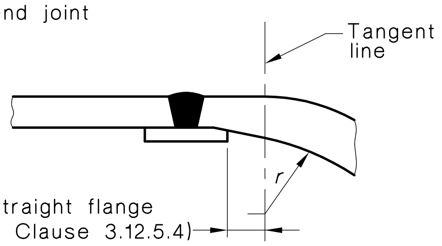
(c) Butt joint—end thicker than shell; median plane approximately coincident



(d) Butt joint—end thinner than shell

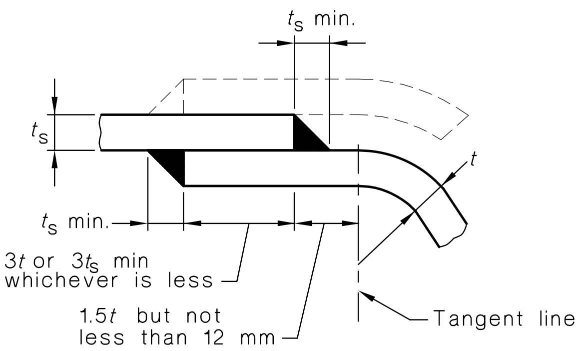


(e) Butt joint—end thinner than shell (alternative to (d))

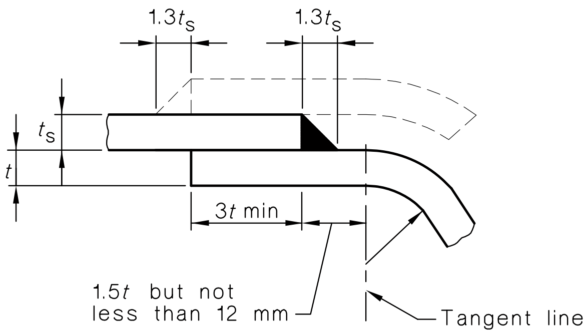


(f) Butt joint—with backing strip

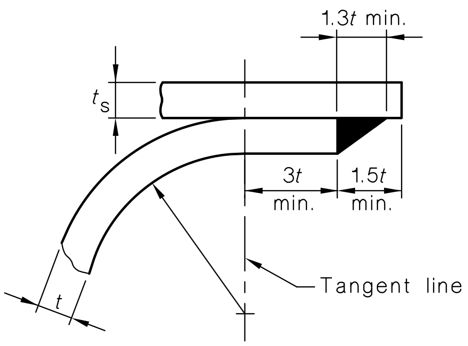
FIGURE 3.12.6 (in part) ATTACHMENT OF DISHED ENDS



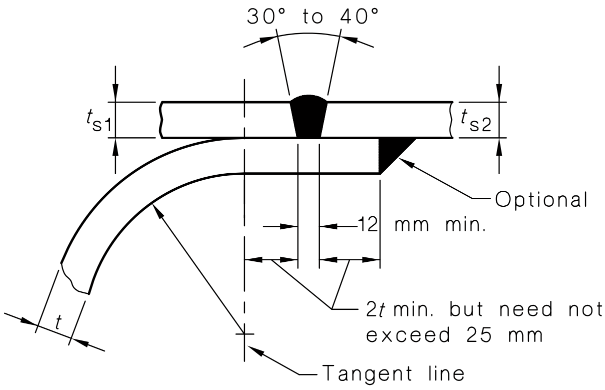
(g) Double fillet lap weld
(Class 3 vessels only)



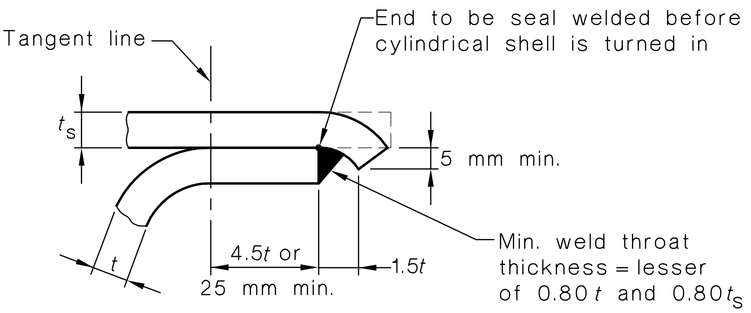
(h) Single fillet lap weld
For limitations see Table 3.5.1.7



(j) Single fillet lap weld
For limitations see Table 3.5.1.7



(k) Intermediate end
(t_{s1} and t_{s2} may be different)



(l) Single fillet lap weld ends convex to pressure
(Class 3 vessels only D > 500mm)

FIGURE 3.12.6 (in part) ATTACHMENT OF DISHED ENDS

3.13 DISHED ENDS SUBJECT TO EXTERNAL PRESSURE

3.13.1 General

Unstayed dished ends of spherical, ellipsoidal or torispherical shape subject to external pressure (i.e. pressure on the convex side) shall be designed in accordance with this Clause (3.13). The thickness so determined shall be not less than that required by Clause 3.4.3. Ends constructed of Group F or Group G steels shall be spherical or ellipsoidal in shape.

3.13.2 Notation

See Clause 3.12.2.

3.13.3 Ellipsoidal ends

The minimum calculated thickness of ellipsoidal ends of seamless or butt joint fabrication at any point after forming shall be the thickness of an equivalent sphere determined in accordance with Clause 3.9.4. The mean diameter shall be taken as equal to the mean diameter of the end multiplied by a factor, F, determined from Equation 3.13.3.

$$F = 0.5\, D_o/h_o + 0.8\, h_o/D_o - 0.4 \qquad \text{. . . 3.13.3}$$

3.13.4 Spherical and torispherical ends

The minimum calculated thickness at any point after form of spherical or torispherical ends shall be the thickness of an equivalent sphere, having a mean diameter equal twice the mean crown radius of the end, determined in accordance with Clause 3.9.4.

3.13.5 Attachment of ends

The required length of straight flange on ends convex to pressure shall comply with the requirements of Clause 3.12.6 or other designs where the crown stress in the end is less than $|0.5f|$, and the design is acceptable to the parties concerned.

3.14 DISHED ENDS—BOLTED SPHERICAL TYPE

3.14.1 General

Circular spherically dished ends with bolting flanges concave or convex to pressure and conforming with Figure 3.14.1 shall be designed in accordance with this Clause (3.14). The thickness so determined shall also comply with Clause 3.4.3.

NOTES:

1 Since $H_r h_r$ in some cases will subtract from the total moment, the moment in the flange ring when the internal pressure is zero may be the determining loading for the flange design.

2 Equations 3.14.3(1) to (8), inclusive, are approximate in that they do not take into account continuity between the flange ring and the dished end. A more exact method of analysis which takes this into account may be used if it meets the requirements of Clause 1.5. Such a method should parallel the method of analysis and allowable stresses for flange design in Clause 3.21. The dished end thickness may be determined by Clauses 3.12 and 3.13 provided that discontinuity stresses are considered.

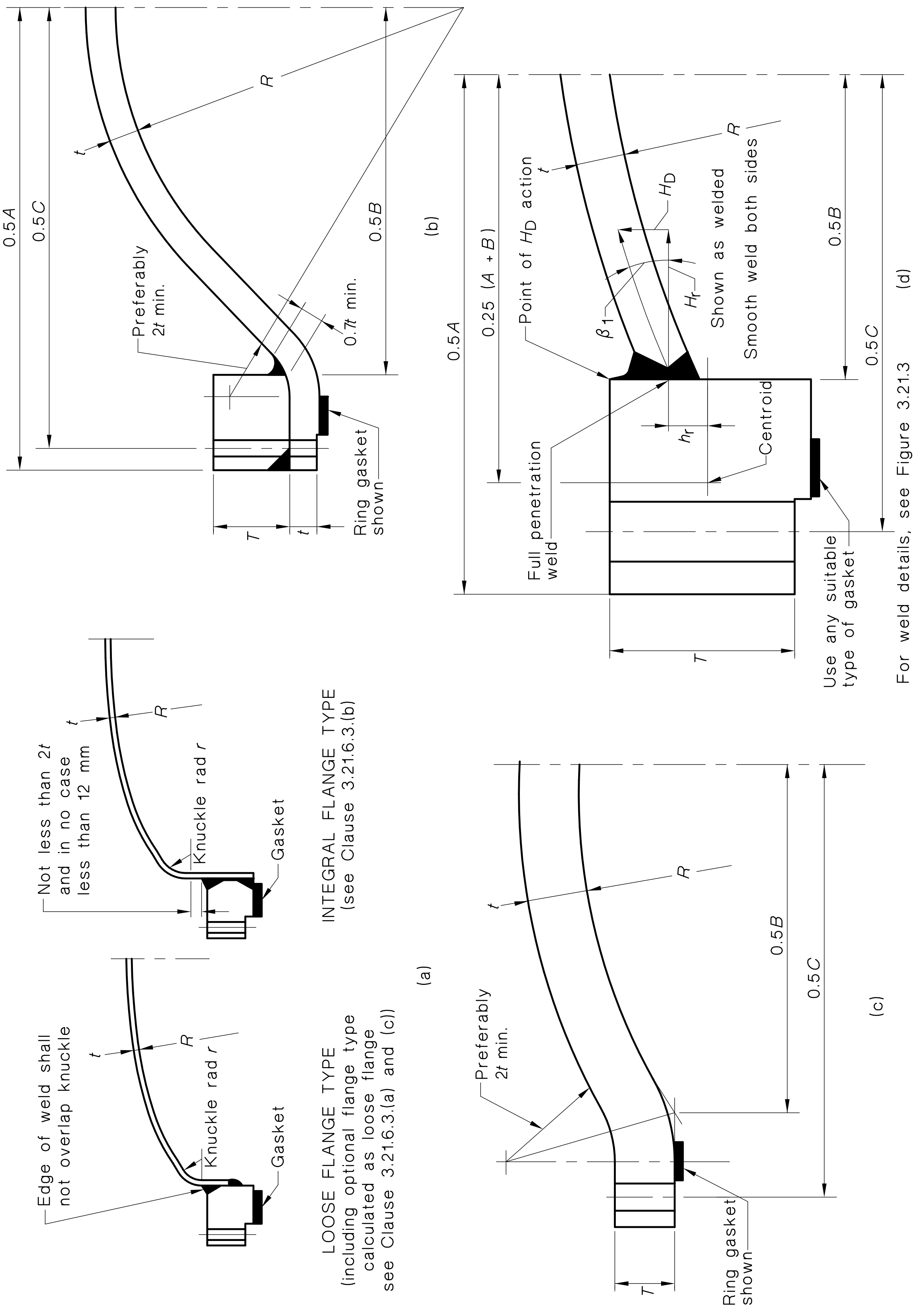


FIGURE 3.14.1 SPHERICALLY DISHED ENDS WITH BOLTING FLANGES

3.14.2 Notation

For the purpose of this Clause (3.14), the following notation applies:

A = outside diameter of flange, in millimetres

B = inside diameter of flange, in millimetres

C = bolt circle diameter, in millimetres

t = minimum calculated thickness of end after forming at the thinnest point (exclusive of added allowances—see Clause 3.4.2), in millimetres

R = inside radius of crown, in millimetres

r = inside knuckle radius, in millimetres

P = calculation pressure (see Clause 3.2.1), in megapascals

= 0 for the gasket seating condition in those equations of Clause 3.14.3 which include the term M_o

f = design tensile strength at the design temperature (see Table B1), in megapascals

T = minimum calculated flange ring thickness, in millimetres

= the greater of the thickness calculated for the operating conditions (P equals calculation pressure) and the gasket seating conditions (P equals 0)

M_o = the total moment determined from Clause 3.21 for both operating and gasket seating conditions, except that for ends in Figure 3.14.1(d), additional moment $H_r h_r$ shall be included (see Note 1 of Clause 3.14.1), in newton millimetres

H_r = radial component of the membrane force in the spherical crown section (equals $H_D \cot \beta_1$), acting at the intersection of the inside of the flange ring with the centreline of the dished end thickness, in newtons

H_D = axial component of the membrane force in the spherical crown section (equals $0.785B^2P$) acting at the inside of the flange ring, in newtons

h_D = radial distance from the bolt circle to the inside of the flange ring, in millimetres

h_r = lever arm of force H_r about centroid of flange ring, in millimetres

β_1 = angle between tangent to the centreline of the dished end at its intersection with the flange ring, and a line perpendicular to the axis of the dished end, in degrees

$$= \arcsin\left(\frac{B}{2R+t}\right)$$

NOTE: $\cot \beta_1 = \left[\left(\frac{2R+t}{B}\right)^2 - 1\right]^{1/2}$

3.14.3 Ends subject to internal pressure (concave to pressure)

The minimum calculated thickness of the end and flange shall be no less than that determined by the following equations:

(a) *Ends of type shown in Figure 3.14.1(a)*

t ≥ thickness determined from appropriate equation in Clause 3.12, and R and r shall not exceed the limits in Clause 3.12.

T ≥ flange thickness determined from Clause 3.21.

(b) *Ends of type shown in Figure 3.14.1(b)* (efficiency of any butt joint in end may be disregarded).

$$t = \frac{5PR}{6f} \qquad \text{. . . 3.14.3(1)}$$

$$T \text{ (for ring gasket)} = \left[\frac{M_o}{fB}\left(\frac{A+B}{A-B}\right)\right]^{1/2} \qquad \text{. . . 3.14.3(2)}$$

$$T \text{ (for full face gasket)} = 0.6\left(\frac{P}{f}\left[\frac{B\,(A+B)\,(C-B)}{A-B}\right]\right)^{1/2} \qquad \text{. . . 3.14.3(3)}$$

NOTE: The radial components of the membrane load in the spherical section are assumed to be resisted by its flange.

(c) *Ends of type shown in Figure 3.14.1(c)* (efficiency of any butt joint in end may be disregarded).

$$t = \frac{5PR}{6f} \qquad \text{. . . 3.14.3(4)}$$

$$T \text{ (for ring gasket)} = Q\left[1+\left(1+\frac{7.5M_o}{PQBR}\right)^{1/2}\right] \qquad \text{. . . 3.14.3(5)}$$

$$T \text{ (for full face gasket)} = Q\left[1+\left[1+\frac{3B(C-B)}{QR}\right]^{1/2}\right] \qquad \text{. . . 3.14.3(6)}$$

where

$$Q = \frac{PR}{4f}\left[\frac{1}{1+6\left(\dfrac{C-B}{C+B}\right)}\right] \text{ for round bolting holes.}$$

$$Q = \frac{PR}{4f}\left[\frac{1}{1+2\left(\dfrac{C-B}{C+B}\right)}\right] \text{ for bolting holes slotted through edge.}$$

(d) *Ends of type shown in Figure 3.14.1(d)* (efficiency of any butt joint in end may be disregarded).

$$t = \frac{5PR}{6f} \qquad \ldots 3.14.3(7)$$

$$T = F + \left(F^2 + Z\right)^{1/2} \qquad \ldots 3.14.3(8)$$

where

$$F = \frac{PB\left(4R^2 - B^2\right)^{1/2}}{8f\left(A - B\right)}$$

$$Z = \frac{M_o\left(A + B\right)}{fB\left(A - B\right)}$$

3.14.4 Ends subject to external pressure (convex to pressure)

Circular spherically dished ends convex to pressure shall be designed to the equations in Clause 3.14.3. The spherical sections shall then be thickened, where necessary, to meet the requirements of Clause 3.13.

3.15 UNSTAYED FLAT ENDS AND COVERS

3.15.1 General

Unstayed flat ends, covers, plates and blind flanges shall be designed in accordance with this Clause (3.15). The thickness so determined shall be not less than that required by Clause 3.4.3. These requirements apply to both circular or non-circular ends and covers. Clause 3.15.5 gives requirements for internally fitted doors.

Some acceptable types of flat ends and covers are shown in Figure 3.15.1.

3.15.2 Notation

For the purpose of this Clause (3.15), the following notation applies.

K = a factor depending upon the method of attachment of end, shell dimensions, and other items as listed below (see Figure 3.15.1)

D_1 = long span of non-circular ends or covers measured perpendicular to short span, in millimetres

D = diameter, measured as indicated in Figure 3.15.1, in millimetres, or short span of non-circular ends or covers measured perpendicular to long span, in millimetres

G = used in the narrow-face calculations to which the flat cover is attached (see Clause 3.21.6).

h_G = gasket moment arm taken from values in the design of the flange to which the plate is attached (refer Clause 3.21.6) or if the flange is not designed, equal to the radial distance from the centreline of the bolts to the line of the gasket reaction, as shown in Figure 3.15.1 (k) and (l), in millimetres

L = perimeter of non-circular bolted end measured along the centre of the bolt holes, in millimetres

l = length of flange of flanged ends, measured from the tangent line of knuckle, as indicated in Figure 3.15.1(a) and (c), in millimetres

m = the ratio $\frac{t_r}{t_s}$

P = calculation pressure, in megapascals

r = inside corner radius on an end formed by flanging or forging, in millimetres

f = design tensile strength at design temperature (see Table B1), in megapascals

t = minimum calculated thickness of flat end or cover (exclusive of added allowances—see Clause 3.4.2), in millimetres

t_e = minimum distance from bevelled end of vessel, before welding, to outer face of end, as indicated in Figure 3.15.1(h) and (j), in millimetres

t_f = actual thickness of the flange on a forged end, at the large end (exclusive of added allowances—see Clause 3.4.2) as indicated in Figure 3.15.1, in millimetres

t_h = actual thickness of flat end or cover (exclusive of added allowances–see Clause 3.4.2), in millimetres

t_r = required thickness of seamless shell, for pressure, in millimetres

t_s = actual thickness of shell (minus any allowances, see Clause 3.4.2), in millimetres

t_w = thickness through the weld joining the edge of an end to the inside of a vessel, as indicated in Figure 3.15.1(g), in millimetres

t_1 = throat dimension of the closure weld, as indicated in Figure 3.15.1(r), in millimetres.

W = total bolt force given for circular ends in Equations 3.21.6.4.4(1) and 3.21.6.4.4(2)

Z = a factor for non-circular ends and covers that depends on the ratio of short span to long span, as given in Clause 3.15.4 (dimensionless)

η = lowest efficiency of any type A (longitudinal) welded joint in an end

3.15.3 Minimum calculated thickness for circular ends

The minimum calculated thickness shall be determined by the following equations:

$$t = D\left(\frac{P}{Kf\eta}\right)^{0.5} \qquad \ldots 3.15.3(1)$$

except for bolted ends, covers and blind flanges with edge moment (see Figure 3.15.1(k) and (l))—

$$t = D\left(\frac{P}{Kf\eta}+\frac{1.78\,Wh_G}{f\eta\,D^3}\right)^{0.5} \qquad \ldots 3.15.3(2)$$

In the latter equation, t shall be the greater thickness calculated for both the operation and gasket seating conditions. For operating condition, P = calculation pressure, f = design strength at calculation temperature, and W is obtained for the operating condition in Clause 3.21. For gasket seating, P = 0, f = design strength at atmospheric temperature, and W is obtained for the gasket seating condition in Clause 3.21.

3.15.4 Minimum calculated thickness for non-circular ends

The minimum calculated thickness of rectangular, elliptical and obround (see Clause 3.18.3) ends and covers shall be determined by the following equations:

$$t = D\left(\frac{ZP}{Kf\eta}\right)^{0.5} \quad \text{. . . 3.15.4(1)}$$

where

$$Z = 3.4 - \frac{2.4D}{D_1} \text{ but not greater than 2.5.}$$

For bolted flanges with edge moment (see Figure 3.15.1(k) and (l)) the thickness shall be determined in the same way as for bolted flanges in Clause 3.15.3, using the following equation:

$$t = D\left(\frac{ZP}{Kf\eta} + \frac{6Wh_G}{f\eta LD^2}\right)^{0.5} \quad \text{. . . 3.15.4(2)}$$

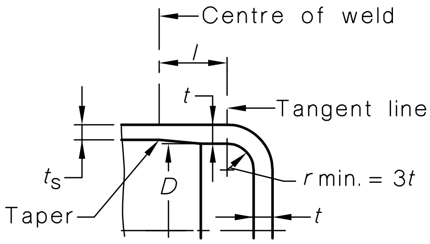
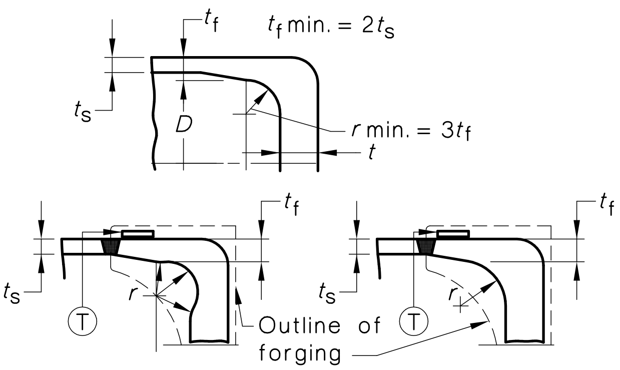
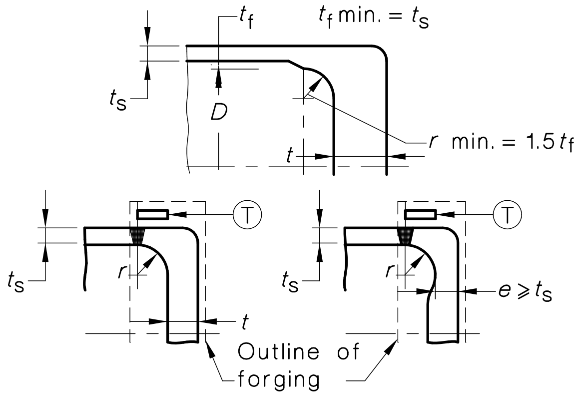
Figure	Value of K	Circular or non-circular ends	Conditions (in addition to those shown in Figure)
(a) 	5.9	Circular or non-circular	No special requirement for l
A1	10.0	Circular only	$l \geq \left(1.1 - 0.8\frac{t_s^2}{t_h^2}\right)\sqrt{(Dt_h)}$ For taper see Clause 3.5.1.8
	10.0	Circular	l less than above; but $t_s = 1.12t_h\sqrt{\left(1.1 - \frac{1}{Dt_h}\right)}$ For length $\geq 2\sqrt{Dt_s}$; For taper see Clause 3.5.1.8
(b-1) 	5.9	Circular or non-circular	Seamless or welded Two welded alternatives machined from forgings (not plate) are shown
(b-2) 	$3/m$, but not more than 5.0	Circular or non-circular	Seamless or welded Two welded alternatives machined from forgings (not plate) are shown

FIGURE 3.15.1 (in part) SOME ACCEPTABLE TYPES OF UNSTAYED FLAT ENDS AND COVERS

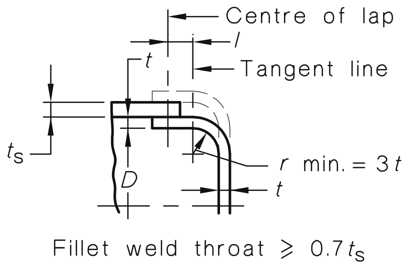
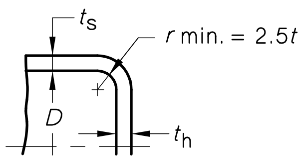
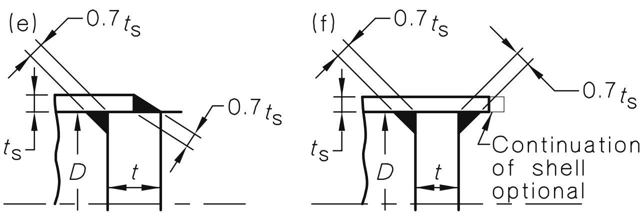
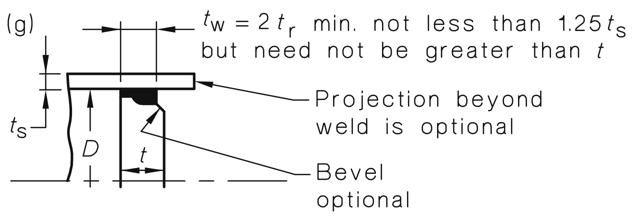
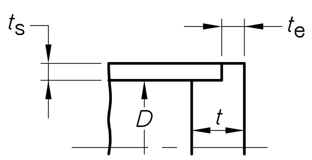
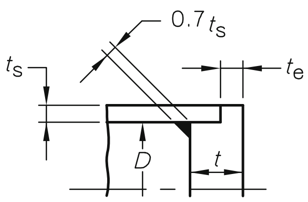
Figure	Value of K	Circular or non-circular ends	Conditions (in addition to those shown in Figure)
(c) 	7.7	Circular	Lap welded or brazed $l \geq$ that given in (a) above
	5.0	Circular or non-circular	Lap welded or brazed. No limit on l
	3.3	Circular	Screwed cap over end. Threads designed against failure by shear, tension or compression resulting from end pressure force using a factor of safety of 4. Threaded parts also at least as strong as the threads for standard piping of the same diameter. Seal welding may be used if desired.
(d) 	7.7	Circular	Integral end by upsetting or spinning of shell as in closing header ends $D \leq 600\text{mm},\ t_h \geq t_s;\ 0.05 \leq \frac{t_h}{D} < 0.25$
(e), (f) 	3/m but not more than 5.0 for circular ends		t_s shall be maintained inwardly for a distance $2\sqrt{Dt_s}$ from the inside face of the end
	3.0 for non-circular ends		Standard single bevel or J weld preparation shall be used
(h) 	3.0	Circular	See Figure 3.17.12(a)-(m) inclusive for details of welded joint
(j) 	3/m but $\leq$ 5.0	Circular	The fillet weld may contribute an amount equal to t_s to the sum of weld dimensions (see Figure 3.17.12(a) to (m) inclusive for outside weld)

FIGURE 3.15.1 (in part) SOME ACCEPTABLE TYPES OF UNSTAYED FLAT ENDS AND COVERS

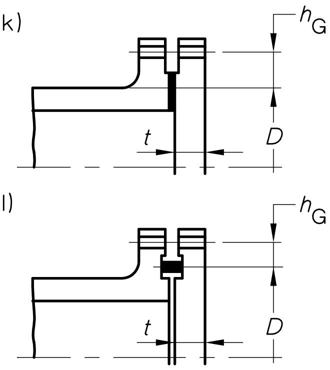
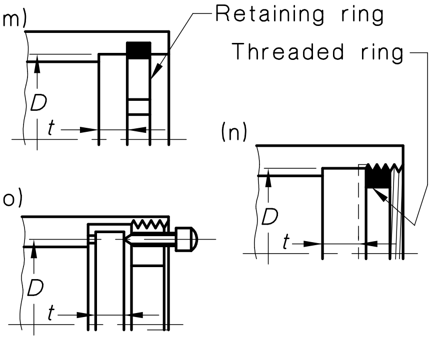
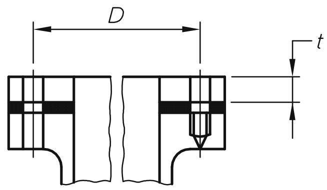
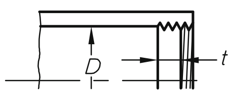
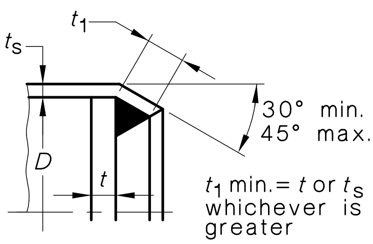
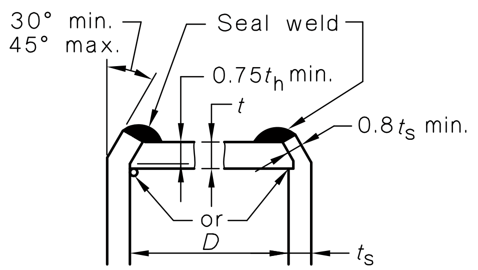
Figure	Value of K	Circular or non-circular ends	Conditions (in addition to those shown in Figure)
(k) (l) 	3.3	Circular and non-circular	Use Equations 3.15.3(2) and 3.15.4(2). The net end plate thickness under the groove or between the groove and the outer edge of the end, shall be not less than: $D\sqrt{\left(\frac{1.78\,Wh_g}{fD^3}\right)}$ (for circular ends and covers) $D\sqrt{\left(\frac{6\,Wh_G}{fLD^2}\right)}$ (for non-circular ends and covers)
(m) (n) (o) 	3.3	Circular	Positive mechanical locking required. All possible means of failure, (shear, tension, compression or radial deformation, including flaring, resulting from pressure and differential thermal expansion ratchetting) are to be designed with a factor of safety of at least 2 on R_{eT} and 4 on R_{mT}. Seal welding may be used, if desired.
(p) 	4.0	Circular and non-circular	Full face joint
(q) 	1.3	Circular	$D \leq 315$ mm. Also applies to ends with integral flange screwed over the vessel end, when thread is designed as for (n). Seal welding may be used, if desired
(r) 	3.0	Circular	$D \leq 450$ mm. Crimping may be done cold when this operation will not injure the metal

FIGURE 3.15.1 (in part) SOME ACCEPTABLE TYPES OF UNSTAYED FLAT ENDS AND COVERS

Figure	Value of K	Circular or non-circular ends	Conditions (in addition to those shown in Figure)
(s) 	3.0	Circular	$D \le 450$ mm. Undercutting for seating shall leave at least 0.80t_s. Bevelling shall be not less than 0.75t_h. Crimping shall be done when the entire circumference of the cylinder is uniformly heated to forging temperature. Also $0.05 \le \frac{t_s}{D} > \frac{P}{f}$ $P \le \frac{f}{5D}$

NOTES:

1 All welding to comply with appropriate requirements of Section 4.

2 In items (b-1) and (b-2), T indicates position of tension test specimen taken from forging.

3 Ends and covers in Group F or Group G steels that are attached by welding shall be attached with full penetration butt welds.

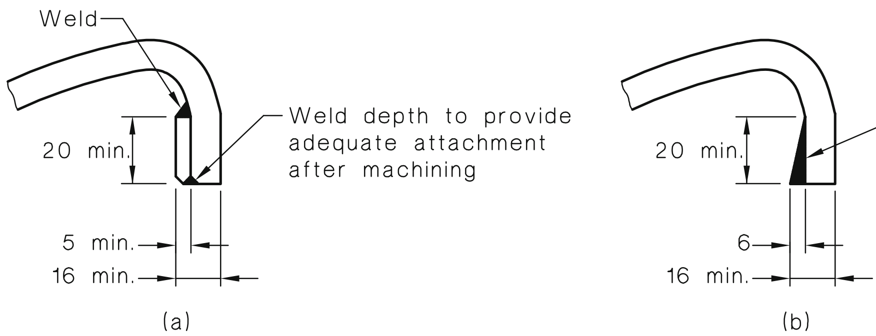
FIGURE 3.15.1 (in part) SOME ACCEPTABLE TYPES OF UNSTAYED FLAT ENDS AND COVERS

3.15.5 Internally fitted doors

3.15.5.1 *General*

Internally fitted flat elliptical or circular doors or elliptical doors formed to the curvature of the surface to which they are fitted shall be secured by studs or bolts and bridge pieces. The doors shall be of plate, built-up or pressed to shape (e.g. stiffened by forming) and subsequently heat treated, or made of one thickness of plate with a machined spigot or recess or other means of securing the jointing material. Doors shall fit closely and properly to the internal surfaces and when the spigot or recess is in a central position it shall not have a clearance greater than 1.5 mm at any point. Doors shall be machined on the bearing surfaces for nuts or collars. The plate shall be inspected before welding and shall be free from material defects.

The width of the gasket bearing surface for an internal manhole door where internal pressure forces the door against a flat gasket shall be not less than 16 mm. Where the plate thickness at the joint is less than 16 mm, the bearing width may be increased by either of the methods shown in Figure 3.15.5.1.



DIMENSIONS IN MILLIMETRES

FIGURE 3.15.5.1 INCREASED WIDTH OF JOINTING SURFACE

3.15.5.2 *Thickness of one-plate doors*

The minimum calculated thickness of an unstiffened one-plate door, either flat or formed to a cylindrical curvature, shall be no less than that determined by the following Equation:

$$t = \left(\frac{K_1 P d^2 + K_2 W_1}{f}\right)^{0.5} \qquad \ldots 3.15.5.2$$

where

t = minimum calculated thickness of one-plate door (exclusive of added allowances, see Clause 3.4.2), in millimetres

P = calculation pressure of vessel to which door is fitted, in megapascals

d = for elliptical doors—minor axis of the opening to which the door is fitted, in millimetres

= for circular doors—diameter of opening to which the door is fitted, in millimetres

W_1 = full load capacity of one stud (core area × design strength for stud material), in newtons

f = design strength at the design temperature (see Table B1), in megapascals

K_1 = stress factor

= 0.40 for all flat doors and for curved doors smaller than 180 mm × 125 mm (see Table 3.15.5.2 for curved doors 180 mm × 125 mm and larger)

K_2 = stress factor

= 0.80 for flat doors and 0.60 for curved doors

D = (diametrical) cylindrical curvature of door, in millimetres

TABLE 3.15.5.2

STRESS FACTOR K_1 FOR CYLINDRICALLY CURVED ELLIPTICAL DOORS FITTED INTERNALLY

Size of opening mm	For values of $t \times D \times 10^{-4}$, mm^2													
	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0
(A) MINOR AXIS OF DOOR PARALLEL TO VESSEL AXIS														
400 × 300	0.07	0.07	0.13	0.18	0.21	0.24	0.26	0.28	0.29	0.31	0.33	0.34	0.35	0.36
380 × 280	0.08	0.08	0.14	0.19	0.24	0.25	0.27	0.29	0.31	0.32	0.34	0.35	0.36	0.37
280 × 180	0.10	0.19	0.24	0.29	0.32	0.34	0.35	0.36	0.36	0.37	0.38	0.38	0.39	0.39
225 × 180	0.16	0.25	0.30	0.33	0.35	0.36	0.37	0.38	0.38	0.38	0.38	0.38	0.39	0.39
180 × 125	0.21	0.31	0.34	0.37	0.37	0.37	0.8	0.38	0.38	0.38	0.38	0.39	0.39	0.40
(B) MAJOR AXIS OF DOOR PARALLEL TO VESSEL AXIS (See Clause 3.18.5.3)														
400 × 300	0.09	0.13	0.22	0.28	0.32	0.34	0.36	0.37	0.38	0.38	0.39	0.40	0.40	0.40
380 × 280	0.10	0.16	0.24	0.30	0.33	0.35	0.37	0.38	0.38	0.39	0.40	0.40	0.40	0.40
280 × 180	0.16	0.34	0.35	0.37	0.38	0.39	0.39	0.40	0.40	0.40	0.40	0.40	0.40	0.40
225 × 180	0.20	0.38	0.38	0.39	0.39	0.39	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40
180 × 125	0.32	0.38	0.39	0.39	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40

3.15.5.3 *Thickness of two-plate doors*

Where a door is made of two plates attached together in such a manner as to adequately resist the shearing force between them, the door may be considered equivalent to a one-plate door. Otherwise, the thickness of the inner plate shall be no less than the minimum calculated thickness of the shell to which the door is fitted and the minimum calculated thickness of the door shall satisfy the relationship shown in the following equation:

$$t = \sqrt{\left(t_1^2 + t_2^2\right)} \qquad \text{. . . 3.15.5.3}$$

where

t = minimum calculated thickness of a one-plate door from Equation 3.15.5.2

t_1 = thickness of the inner plate, in millimetres

t_2 = thickness of the outer plate, in millimetres.

3.15.5.4 *Bolting*

Studs, bolts, nuts and washers shall comply with Clause 3.21.5.4, except that the minimum size shall be 16 mm.

A door to fit an opening not larger than 225 mm × 180 mm elliptical or 180 mm diameter may be fitted with one stud or bolt. A door for a larger opening shall be fitted with a minimum of two studs or bolts unless otherwise agreed between the parties concerned. Where studs are located at or near the focal points of the ellipse, only one stud load (W_1) may be used in Equation 3.15.5.2.

The number and size of studs shall provide an adequate gasket seating force (W_{m2}) in accordance with Equation 3.15.5.4(1):

$$W_{m2} = bG_1y \qquad \text{. . . 3.15.5.4(1)}$$

where

W_{m2} = minimum required bolt force for gasket seating, in newtons

b = effective gasket width (see Table 3.21.6.4(B)), in millimetres

G_1 = length of gasket periphery at the mid-point of the contact surface (3.14 times diameter for circular door), in millimetres

y = gasket seating stress from Table 3.21.6.4(A), in megapascals.

NOTE: For high pressures, the gasket might need to be of metal, or metal clad, to resist the internal pressure in the operating condition.

Each stud or bolt shall be fixed to the door by one of the following methods:

(a) A stud with an integral collar machined on the bearing face, which is fitted through the door and riveted over on the inside. The rivet head shall be of standard dimensions and there shall be a countersink under the head.

(b) A stud with an integral collar machined on the bearing face, which is screwed through the door, and is either substantially riveted over on the inside, or is fitted with a nut on the inside, or is fillet-welded on the inside.

(c) A stud that is screwed into a door thicker than the stud diameter, by at least one full stud diameter, and is lock-welded. Where it is essential to unscrew the stud to permit removal of the door, means shall be provided for separately locking the stud rather than lock-welding.

(d) A stud that is forged solid with a door not exceeding 150 × 100 mm.

(e) A stud that is welded as shown in Figure 3.15.5.4(c) and (d).

(f) A bolt that is fitted through the door with the head inside, and seal-welded as shown in Figure 3.15.5.4(a) and (b).

(g) A square-headed bolt that is used in substantial T-slotted lugs securely fastened or welded to the door. Such bolts and slots shall be machine-finished.

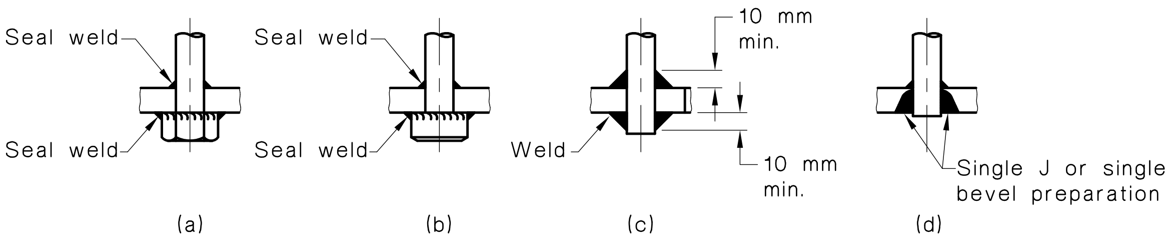


FIGURE 3.15.5.4 ACCEPTABLE METHODS OF FIXING STUDS ON MANHOLE OR HANDLEHOLE DOORS

3.15.5.5 *Bridges or dogs*

Bridges or dogs shall be constructed by one of the following methods:

(a) Forged.

(b) Pressed.

(c) Flame cut to shape.

(d) Cast from material complying with AS 1565 alloy C 86300, provided the vessel does not exceed 400 mm diameter, nor operate at a pressure exceeding 240 kPa.

(e) Fabricated by welding.

The maximum stress calculated as a simple beam of length equal to the centre distance of the bridge supports shall not exceed the stress stated in Table B1 based on the force (W_1) referred to in Clause 3.15.5.2.

NOTE: In the design of an internally fitted door involving bridges or dogs, it is intended that the bridges or dogs will yield before the studs or the door.

3.16 STAYED FLAT ENDS AND SURFACES

3.16.1 General

Stayed and braced flat ends and other surfaces shall be designed in accordance with the requirements of this Clause (3.16). The minimum calculated thickness shall be increased where necessary to carry the additional loads enumerated in Clause 3.2.3 and to meet the requirements of Clauses 3.4.2 and 3.4.3.

Stayed flat ends of Class 1H and 2H construction are frequently subject to internal force and deflection and therefore require a detailed analysis for assessment of the working stresses. For proven simple vessels that are not subject to large temperature differentials, stayed ends may be designed in accordance with Clause 3.16 with values for design strength from Table B1(A).

Figure 3.16.1 shows some acceptable methods of staying surfaces. Other methods may be used where they provide equivalent safety and performance.

Where leakage past a stay would be dangerous, as in certain chemical processes, the plate shall not be perforated for supporting stays. Stays of the types shown in Figure 3.16.1(b), (c), (d) or (e) shall be used where the process face of a jacketed vessel is provided with corrosion-resistant linings except where an alternative configuration provides similar strength and corrosion resistance.

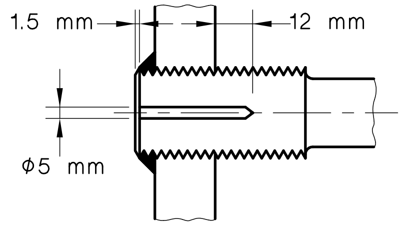
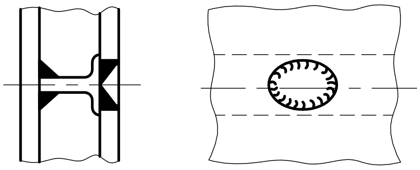
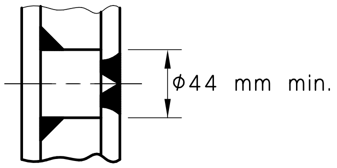
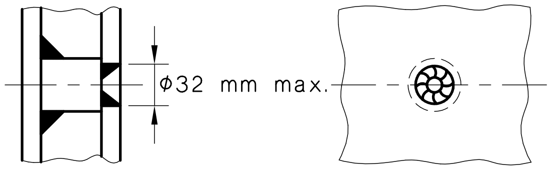
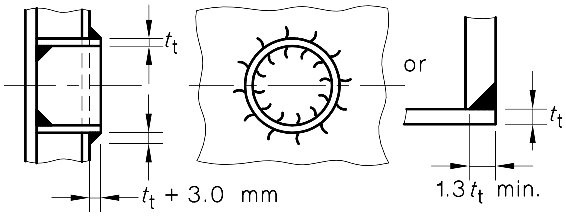
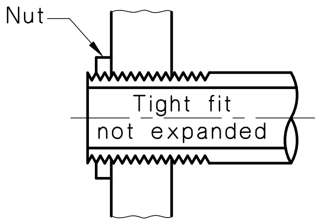
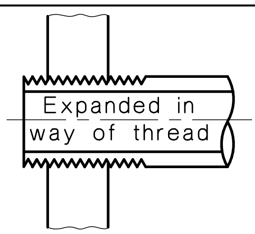
Figure	Type	*K* value	Name	Remarks
(a)		3.75	Staybar: screwed and welded (or riveted)	The staybar may be riveted to form substantial head Middle portion of stay should be reduced to the root diameter of the thread
(b)		4.0	Stayblock: welded	2.1 MPa max. Plate thickness ≤ 12 mm. Fillet welds ≤ plate thickness. Inside welds shall be properly inspected before closing plate is attached. Spacing of stays determined by Clause 3.16.4.
(c)		4.5	Stayblock: welded	
(d)		4.0	Stayblock: welded	
(e)		5.5	Staytube: welded	Inside diameter of staytube is greater than its length
(f)		5.75	Staytube: screwed and nutted	Tube not less than 5 mm thick at any point. Screwed not finer than 2.3 mm pitch. Stays with plus threads at one end shall be thickened by forging, and annealed. Body of stay may be reduced to bottom of thread.
(g)		4.25	Staytube: screwed	

FIGURE 3.16.1 (in part) SOME ACCEPTABLE METHODS FOR STAYING OF SURFACES

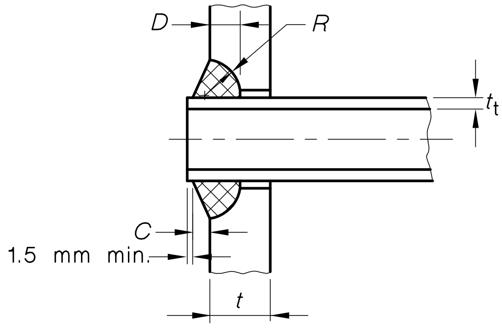
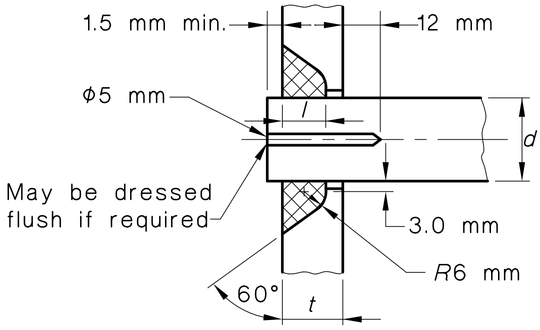
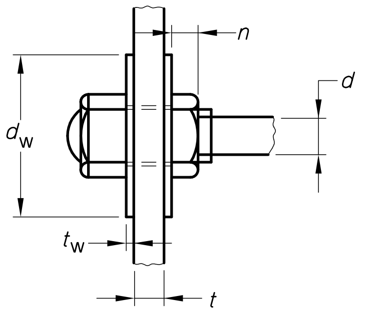
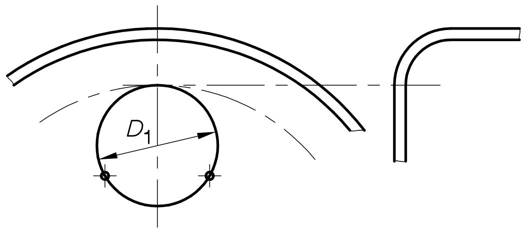
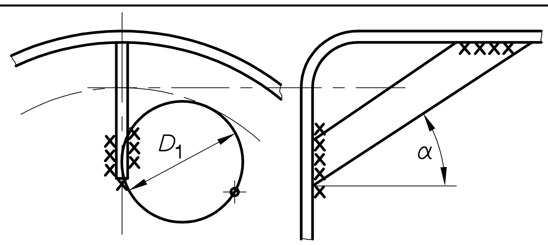
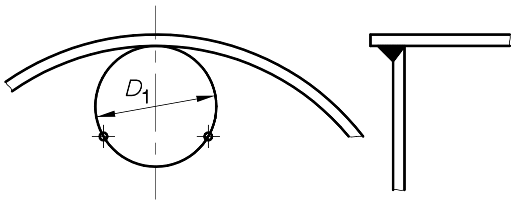
Figure	Type	*K* value	Name	Remarks
(h)		4.25	Staytube: welded	D = t_t min. R = 1 to 1.5 t_t min. C = t_t or 3 mm whichever is greater
(j)		5.05	Staybar: welded	$l = t - 3$ mm; or $0.25d + 3$ mm whichever is the smaller Standard single bevel and J weld are also acceptable
(k)		6.25	Staybar with nuts and small washers	d_w not less than $2.25d$ t_w not less than 6 mm n not less than 0.66 d Inside washer may be omitted. Middle portion of stay should be reduced to the root diameter of the thread
		6.95	Staybar with nuts and large washers	d_w not less than $3.5d$ and not less than $0.3D_1$(see Figure (l)) t_w not less than $0.66t$
(l)		5.0	Flanged support	
(m)		5.0	Welded stay	
(n)		5.5	Flat end welded to shell	

FIGURE 3.16.1 (in part) SOME ACCEPTABLE METHODS FOR STAYING OF SURFACES

3.16.2 Notation

For the purpose of this Clause (3.16), the following notation applies:

t = minimum calculated thickness of stayed plate (exclusive of added allowances, see Clause 3.4.2), in millimetres

P = calculation pressure, in megapascals

f = design strength (see Table B1) at the maximum temperature at which P is operative (allowance shall be made, if necessary, for thermal stresses caused by temperature gradients), in megapascals

A = distance between rows of stays, in millimetres

B = pitch of stays in rows, in millimetres

D_1 = diameter of the largest circle that can be drawn having a circumference passing through at least three points of support without enclosing any other support, with at least one support located on any semi-circular arc of such circle (see Figure 3.16.2), in millimetres

K = a constant depending upon method of attachment of stay to plate and as given in Figure 3.16.1. Where various forms of support are used, the value of K shall be the mean of the three greatest values for support methods used, provided at least one of these supports is located on any semi-circular arc of the circle with diameter D_1

M_g = flat plate margin, in millimetres

$$= 0.9t\left(\frac{f}{P}\right)^{0.5}$$

3.16.3 Plate thickness

The minimum calculated thickness for stayed or braced flat surfaces shall be determined by the following equation:

$$t = \left(\frac{PD_1^2}{fK}\right)^{0.5} \qquad \text{. . .3.16.3}$$

Where tubes are expanded into a stayed plate the thickness of the plate within the tube nest shall be not less than 12 mm where the tube holes do not exceed 50 mm diameter and no less than 15 mm where the tube holes exceed 50 mm diameter.

The minimum thickness of plates to which stays are applied, in other than cylindrical or spherical outer shell plates, shall be 8 mm except for welded construction covered by this Standard.

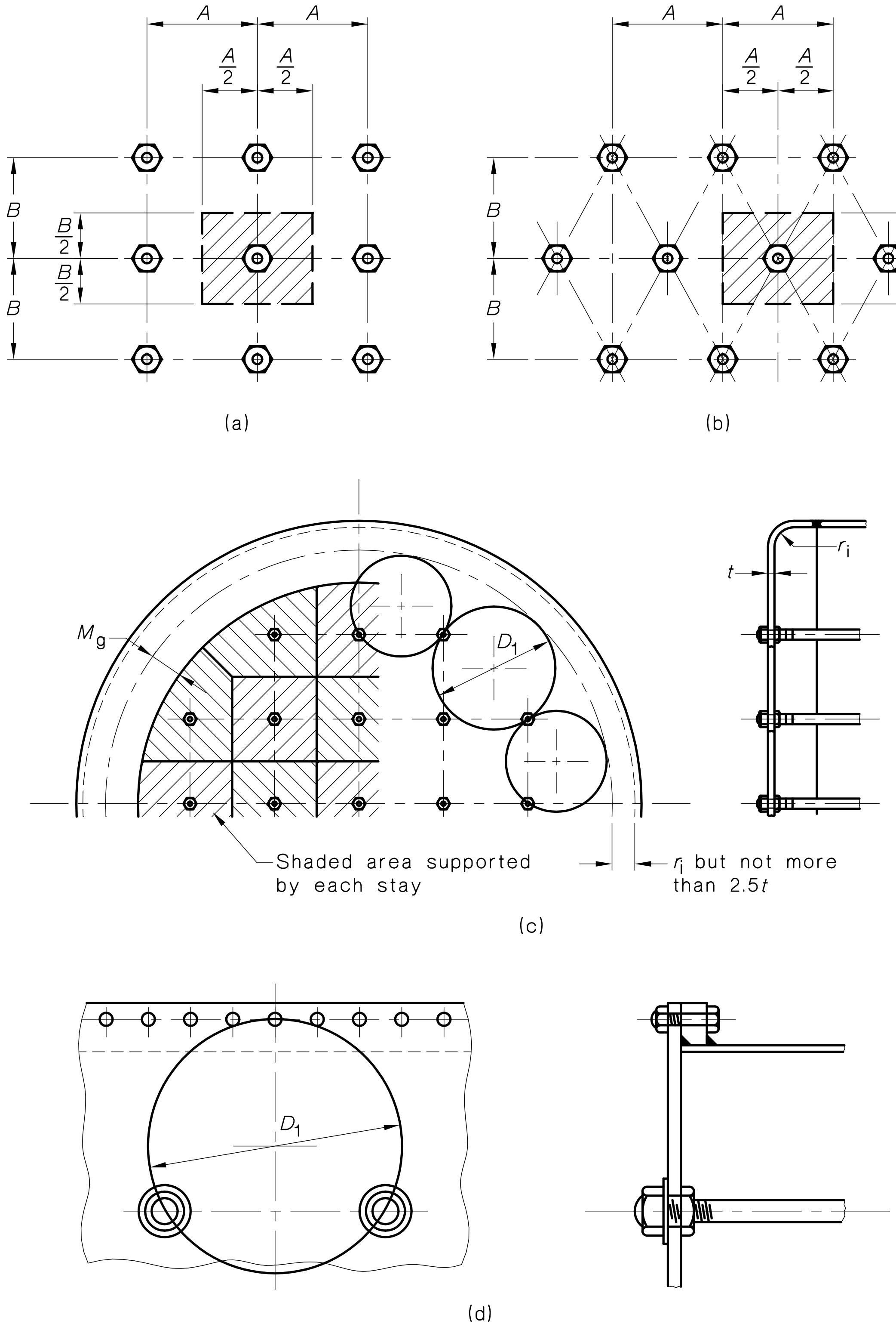
3.16.4 Minimum pitch of staytubes

The centrelines of the tubes, measured at the tubeplate, shall not be closer together than $1.125d + 12$ mm where d is the outside diameter of the tube in millimetres.

3.16.5 Staybars and staytubes

3.16.5.1 *Material*

Each staybar shall be made from rolled bar without weld in its length, except where it is attached to the plate it supports.



NOTE: Gross area supported by one stay is shown shaded. To obtain net areas deduct sectional area of stay.

FIGURE 3.16.2 PITCHES AND AREAS SUPPORTED BY STAYS

3.16.5.2 *Dimensions*

The required area of a staybar or staytube at its minimum cross-sectional area (usually taken at the nominal root of the thread) and exclusive of corrosion allowance, shall be obtained by dividing the load on the stay by the allowable stress for the material (see Table B1), and multiplying the result by 1.10. The load carried by the stay is the product of the area supported by the stay (see Clause 3.16.5.3), the calculation pressure, and the secant of the angle, α, between the longitudinal axis of the vessel and the stay (see Figure 3.16.1(m)).

Where the length of stay exceeds 14 stay diameters, the factor 1.10 above shall be replaced by 1.34.

3.16.5.3 *Area supported by stay*

The area supported by each stay shall be the area enclosed by lines passing midway between the stay and/or adjacent points of support and/or by the flat plate margin as the case may be (see Figure 3.16.2). A deduction may be made for the area occupied by the stay.

3.16.5.4 *Staybars—axial drilling*

It is recommended that, where practicable, screwed staybars less than 350 mm long and all welded staybars, subject to any bending, be drilled axially with a telltale hole (5 mm diameter) to a depth of 12 mm beyond the inner face of the plate (see Figure 3.16.1(a)).

3.16.5.5 *Attachment*

Screwed staybars and staytubes shall, where practicable, be normal to the plate surfaces, but where this is not practicable stays fitted with nuts shall be provided with taper washers to provide a proper bed for the nuts.

Stays screwed through the plates and not normal thereto shall have no less than four engaging threads of which at least two shall be full threads. Holes for screwed stays shall be drilled full size, or punched 6 mm less than full size for plates over 8 mm thick and 3 mm less for plates not exceeding 8 mm thick. After punching, the hole shall be drilled or reamed to full diameter. The holes shall be tapped true with full thread.

The ends of screwed staytubes shall project through the tubeplate by not less than 6 mm and by not more than 10 mm.

Weld dimensions shall be such as will transmit the stay load using a design strength not more than 50% of the design strength, f, for the weaker material in the joint.

For other requirements see Figure 3.16.1.

3.16.5.6 *Staybar supports*

Horizontal longitudinal through staybars over 5 m long shall be supported at or near the middle of their length.

3.16.6 Gusset and other stays

Surfaces may also be stayed with one or more of the following types of stays:

(a) Diagonal staybars.

(b) Diagonal gusset stays.

(c) Diagonal link stays.

(d) Transverse or radial plate stays (or ribs).

(e) Dimpled or embossed plate welded to another like plate or to a plain plate.

Stays of types (a), (b) and (c) above shall be designed in accordance with the requirements of AS 1228, except that fillet welds shall comply with the requirements of Clause 3.5.

Stays of Type (d) above shall be designed as beams supporting a load, which is determined in accordance with Clause 3.16.5.2, and with a bending stress across a solid section not exceeding the lesser of that in Clause 3.3.7 and 75 percent of the specified minimum yield strength. Alternatively, these stays may be designed to meet the requirements of Clause 3.1.3.

Dimpled and embossed plate of Type (e) above shall be designed and manufactured in accordance with the requirements in ASME BPV-VIII-1 for dimpled and embossed assemblies except that—

(i) for resistance welded two-plate assemblies, the maximum thickness of any plain plate shall be 10 mm; and

(ii) where the weld attachment is made by fillet welds around holes or slots, the design shall comply with this Clause (3.16).

3.17 FLAT TUBEPLATES

3.17.1 General

The design of flat tubeplates in tubular heat exchangers shall comply with either AS 3857, ASME BPV-VIII-I-UHX, or EN 13445. The TEMA standards and ISO 16812 may also be used, except to determine tube plate thickness.

3.17.2 Not allocated.

3.17.3 Not allocated.

3.17.4 Not allocated.

3.17.5 Not allocated.

3.17.6 Not allocated.

3.17.7 Not allocated.

3.17.8 Tubeplate ligament

3.17.8.1 *Minimum ligament*

The minimum tubeplate ligament of at least 96% of ligaments shall be not less than the nominal ligament minus the ligament tolerance where—

(a) the nominal ligament is the difference between the nominal pitch and the nominal diameter of the tube holes (see Clauses 3.17.9 and 3.17.10 for tube pitch and tube hole);

(b) the ligament tolerance—

   (i) equals 2 × drill drift tolerance + 0.5 mm for tubes less than 15 mm OD; or

   (ii) equals 2 × drill drift tolerance + 0.8 mm for tubes 15 mm OD and larger;

   where drill drift tolerance equals 0.04 t_p/d_o, and t_p is nominal tubeplate thickness.

The minimum tubeplate ligament for the remaining 4% of ligaments shall be not less than $(p - d_o)/2$, rounded to the lower 0.1 mm.

In assessing minimum ligament width no allowance need be made for anchor grooving 0.5 mm deep or less.

3.17.8.2 *Tubes welded to tubeplate*

Where tubes are to be fixed to the tubeplate by welding, the ligament width shall be sufficient to permit proper execution of welding and to develop the strength of attachment (see Figure 3.17.11), and shall also comply with the requirements in Clause 3.17.8.1.

3.17.9 Tube pitch

Where tubes are fixed by expansion only, the nominal pitch of tubes p shall be no less than the following:

(a) $1.25d_o$ for all exchangers not conforming to Item (b).

(b) $1.17d_o$ for special exchangers where close pitching is required, provided that the consequence of tube joint leakage does not involve any hazard or any unacceptable service difficulties, and that the manufacturer establishes that adequate holding power of the tubes can be maintained.

To ensure the maintenance of tube tightness with reduced ligaments, the following should be met:

(i) The maximum difference between temperatures of shell and tube-side fluids does not exceed 20°C.

(ii) The tubes have ends in a condition that the yield stress of the ends is less than 75% of the yield stress of the plate (this will normally be met by using non-ferrous tubes in steel tubeplates).

(iii) The maximum fluid temperature should not exceed 65°C where the tube and tubeplate have markedly different thermal expansion coefficients.

3.17.10 Tube holes

3.17.10.1 *Diameter and finish*

The tube holes in tubeplates into which tubes are to be expanded shall be machined to a workmanlike finish to comply with the tube hole tolerance and maximum diametral clearance given in Table 3.17.10. The inside edges of tube holes, and the outside edges, where tubes are to be belled or beaded, shall have the sharp edge removed.

Where so specified in Clause 3.17.11.1 tube holes for expanded joints shall be machined with one or more anchor grooves, each approximately 3 mm wide × 0.5 mm deep.

The tube holes in tubeplates into which tubes are to be welded without prior or subsequent expansion may be made by any means, e.g. profile flame cutting, subject to the following:

(a) The maximum diametral clearance shall not exceed—

(i) $\frac{t_t}{2}$; or

(ii) $\frac{d_o}{30}$, whichever is the less,

where

t_t = the wall thickness of tube

d_o = the outside diameter of tube

(b) Significant surface grooves parallel to the hole axis shall be limited to one per tube hole, but shall not extend for more than 50% of the depth of the hole. Such grooves are not permitted within 15° of the minimum ligament between holes. This surface groove shall not exceed the following depth:

0.5 mm for tubes ≤25 mm OD.

1.0 mm for tubes 32 ≤ OD ≤ 50 mm.

1.2 mm for tubes 63 ≤ OD ≤ 75 mm.

1.5 mm for tubes >75 mm OD.

The remainder of the tube hole surface shall have a fine finish. Minor repairs by welding and grinding of grooves exceeding these limits are permitted.

(c) Tube holes for vessels operating at pressures exceeding 2.1 MPa, or temperatures above 175°C or below 0°C, shall be finish machined.

(d) The minimum ligament shall comply with the requirements of Clause 3.17.8.

Tubes of outside diameter and tolerances and nominal hole diameters other than listed in Table 3.17.10, may be used provided—

(i) the maximum diametral clearance (by interpolation if necessary) is not exceeded;

(ii) the nominal tube hole diameter does not exceed the nominal outside diameter of the tube (d_o) by more than 2.5 mm or 0.01 d_o, whichever is greater; or

(iii) as an alternative to Item (ii), a suitably modified value of d_o is used in Clause 3.17.11.4 to allow for any weakening in excess of that permitted for standard tubes.

TABLE 3.17.10

TUBE HOLE DIAMETER, TOLERANCE AND CLEARANCE FOR EXPANDED TUBES

millimetres

Nominal OD of tube d_o	Tolerance on tube OD	Nominal tube hole diameter and tolerance				Maximum nominal diametral clearance (Notes 1 and 2) S		Maximum 'plus' tolerance for 4% of holes
		Standard fit		Special close fit (Note 3)				
		Nom. dia. of hole	Tolerance (Note 1)	Nom. dia. of hole	Tolerance (Note 1)	Standard fit	Special fit	
6.35	±0.1	6.57	+0.05 −0.1	6.53	±0.05	0.38	0.32	0.18
9.525	±0.1	9.75	+0.05 −0.1	9.70	±0.05	0.38	0.32	0.18
12.7	±0.1	12.95	+0.05 −0.1	12.78	±0.05	0.41	0.32	0.2
15.875	±0.1	16.12	+0.05 −0.1	16.07	±0.05	0.41	0.36	0.25
19.05	±0.1	19.31	+0.05 −0.1	19.25	±0.05	0.41	0.36	0.25
25.4	±0.15	25.70	+0.05 −0.1	25.65	±0.05	0.51	0.46	0.25
31.75	±0.15	32.10	+0.075 −0.15	32.03	±0.075	0.58	0.51	0.25
38.1	±0.2	38.56	+0.075 −0.13	38.45	±0.075	0.74	0.64	0.25
50.8	±0.25	51.36	+0.075 −0.18	51.26	±0.075	0.89	0.76	0.25

NOTES:

1 96% of tube holes are to meet this requirement. The remaining 4% may have a plus tolerance on the hole not exceeding the values listed, and a correspondingly increased clearance.

1 Maximum diametral clearance = maximum hole diameter (nominal + tolerance) − minimum tube diameter (nominal − tolerance).

2 This class of fit may be specified by the purchaser to minimize work-hardening and the resultant loss of resistance to stress corrosion, e.g. in austenitic steel tubes.

3.17.10.2 *Location in welded joints*

Tube holes may be machined through double-welded butt joints provided the joint, for a distance of at least three tube hole diameters, on each side of any such hole, complies with the following requirements:

(a) *For Class 1 vessels* The joints shall meet the requirements for longitudinal welds, i.e. 100% radiography or ultrasonic test, and any required postweld heat treatment is carried out before tube welding. The welds shall be machined at both sides and the surfaces, including the hole surface, shall be examined by magnetic particle or penetrant fluid crack detection.

(b) *For Class 2 or 3 vessels* The joints shall meet the requirements for longitudinal welds, and shall be examined by magnetic particle or penetrant fluid crack detection. The appropriate welded joint efficiency shall be included in the ligament efficiency when calculating tubeplate thickness.

3.17.11 Tube-to-tubeplate attachment

3.17.11.1 *General*

Tubes shall be fixed into the tubeplate by one of the following methods:

(a) Expanding, or expanding followed by beading, by belling, by beading and seal welding, or by belling and seal welding.

(b) Seal welding followed by expanding.

(c) Welding with or without expanding.

(d) Ferrule and packing joint.

(e) Screwing and expanding, with or without seal welding.

(f) Other methods proven to be acceptable by long and satisfactory experience or by tests.

For lethal or flammable fluid applications, tube holes for expanded joints shall be machined with two anchor grooves or shall be seal welded, except where the joint has been proven to be acceptable by evidence of long and satisfactory experience or by tests in accordance with Appendix A, ASME BPV VIII-1.

For other applications, where the design pressure is equal to or greater than 2.1 MPa or the temperature is over 175°C, tube holes for expanded joints shall be machined with one anchor groove in tubeplates less that 25 mm thick and with two anchor grooves in tubeplates 25 mm or greater in thickness, except that where the strength of the attachment complies with Clause 3.17.11.4, grooves may be omitted.

Seal welds shall not be considered to contribute to the strength of the attachment but their size shall be sufficient to avoid any cracking. When the weld or heat-affected zone of portion of the tube inside the tubeplate could be subject to significant corrosion, the tube shall also be expanded.

Where tubes are fixed by expanding, the thickness of tubeplates, less any corrosion allowances, shall be not less than 10 mm or $0.125d_o$ + 8.25 mm whichever is less, and should be no less than the recommended thickness given in Table 3.17.11.1. Where tubes are fixed by welding, the plate thickness less any corrosion allowances shall be sufficient to enable the attachment to be made satisfactorily.

TABLE 3.17.11.1

RECOMMENDED MINIMUM TUBEPLATE THICKNESS FOR EXPANDED TUBES

Tube outside diameter, d_o mm	Minimum tubeplate thickness, t_p, min. mm
12	12
15	12
20	15
25	20
32	22
38	25
50	32

3.17.11.2 *Not allocated*

3.17.11.3 *Expansion of tubes*

Where tubes are fixed by expansion only, the tube ends shall be flush with or may project beyond the face of the tubeplate. For non-toxic, non-corrosive applications at design pressures not exceeding 3 MPa and tubes not exceeding 20 mm outside diameter, tubes may be recessed not more than 3 mm below the tubeplate surface provided the strength of the attachment complies with the requirements of Clause 3.17.11.4.

3.17.11.4 *Strength of attachment of tube*

Where tubes are welded into the tubeplate, the calculated load on the tube shall not exceed—

$$\pi d_o \times \text{ throat of weld} \times 0.8f \qquad \ldots 3.17.11$$

where

d_o = nominal outside diameter of tube, in millimetres

f = design strength for tube or plate, whichever is the lesser, in megapascals.

Where tubes are fixed by expanding into the tubeplate, the strength of the attachment shall be established by experience with successful exchangers or by actual tests. A safety factor of four shall be used on loads determined by test.

NOTE: For further guidance to allowable tube to tubeplate joint loads refer to Appendix A, ASME BPV VIII-1.

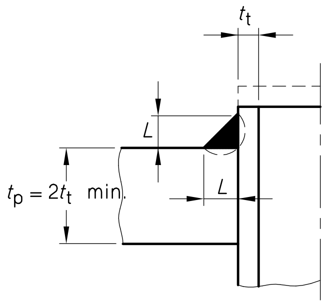
3.17.11.5 *Welded attachment of tubes*

Where tubes are welded into the tubeplate the welding procedure shall be qualified by suitable tests on simulated joints to ensure adequate penetration and freedom from unacceptable defects. Typical weld preparations and sizes are given in Figure 3.17.11.

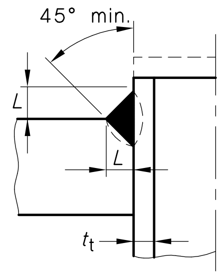
NOTE: Light expanding of the tube is recommended after welding, except with austenitic Cr-Ni steel tubes which should be welded without any expansion, as cold work may affect the corrosion performance of these steels.

3.17.12 Attachment of tubeplate to shell

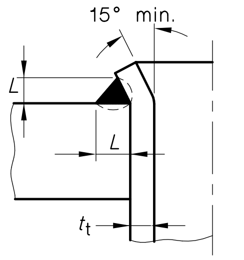
The attachment shall be in accordance with Figure 3.17.12 or other methods providing equivalent performance and safety.



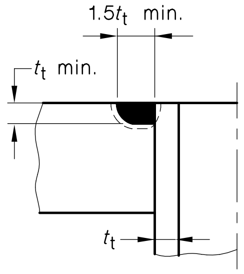
(a) Fillet welded
(Note 2)



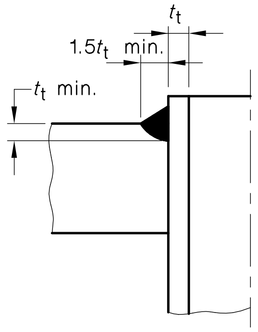
(b) Bevel-fillet welded



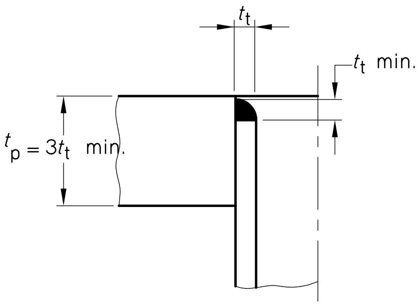
(c) Flared-fillet welded



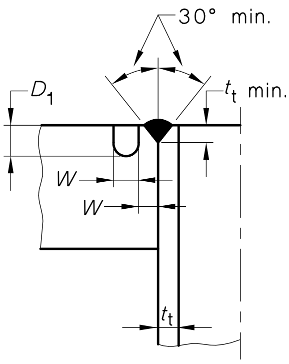
(d) Flush-J welded
(Note 3)



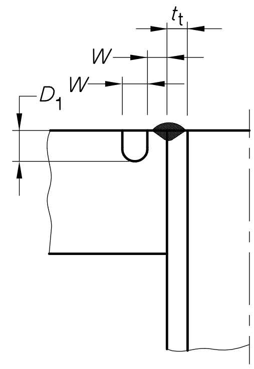
(e) J-fillet welded (Note 3)



(f) Inside fillet welded (Note 9)
(t_t = 3 mm min. for manual metal arc welding)



(g) Grooved-V welded (Notes 4,7)
(use when $t_t \geqslant 5$ mm)



(h) Grooved-edge welded (Notes 4,7,8)
(use when $t_t \leqslant 5$ mm)

NOTES:

1 t_t = nominal thickness of tube
t_p = nominal thickness of tubeplate
$L \geq t_t$
D_1 = $1.5t_t$ to $2.0\ t_t$
$W = t_t$

2 Minimum distance between tubes = 2.5 t_t or 8 mm, whichever is less.

3 This preparation is preferred when there is danger of burning through tube.

4 Tubeplate should be examined for laminations before machining.

5 When conditions are onerous preference should be given to Figure (d) or (e).

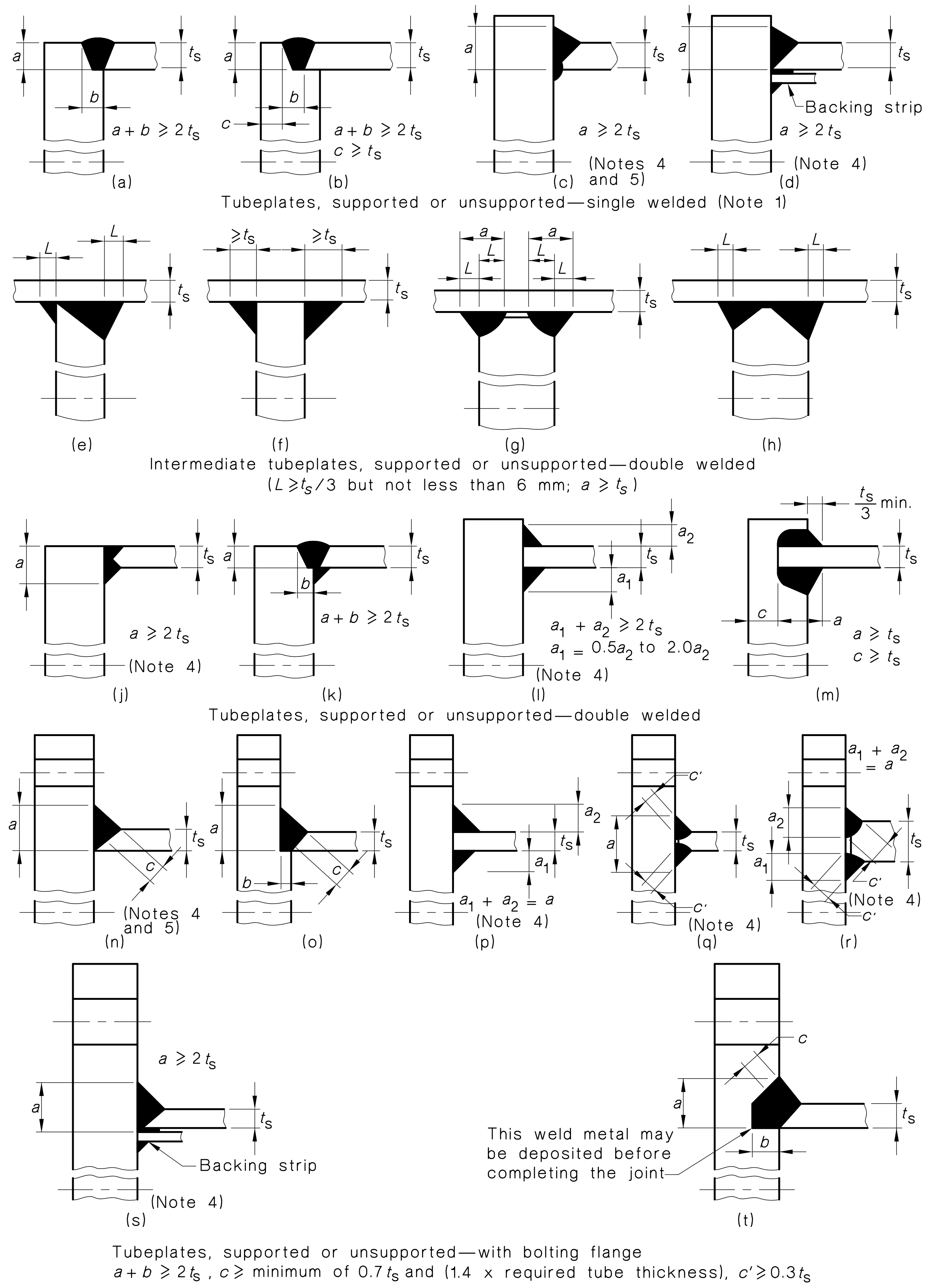
6 See Clauses 3.17.11 for holes and tube expansion.

7 This preparation is preferred where minimum plate distortion is required.

8 This preparation is not suitable for metal-arc welding. Use filler rod where t_t > 1.5 mm (acetylene), or > 2.5 mm (GTAW).

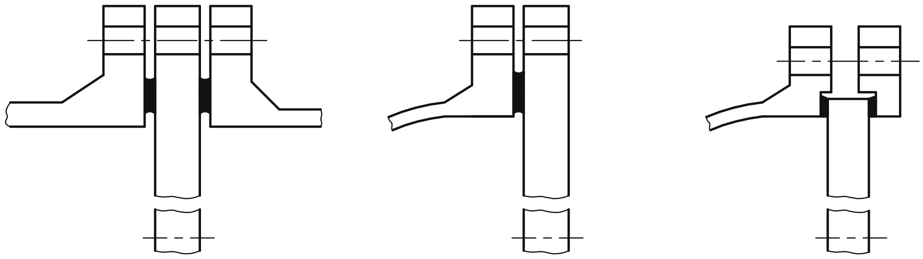
9 This preparation is permissible for low pressure and small temperature fluctuations.

FIGURE 3.17.11 TYPICAL WELDED ATTACHMENT OF TUBES



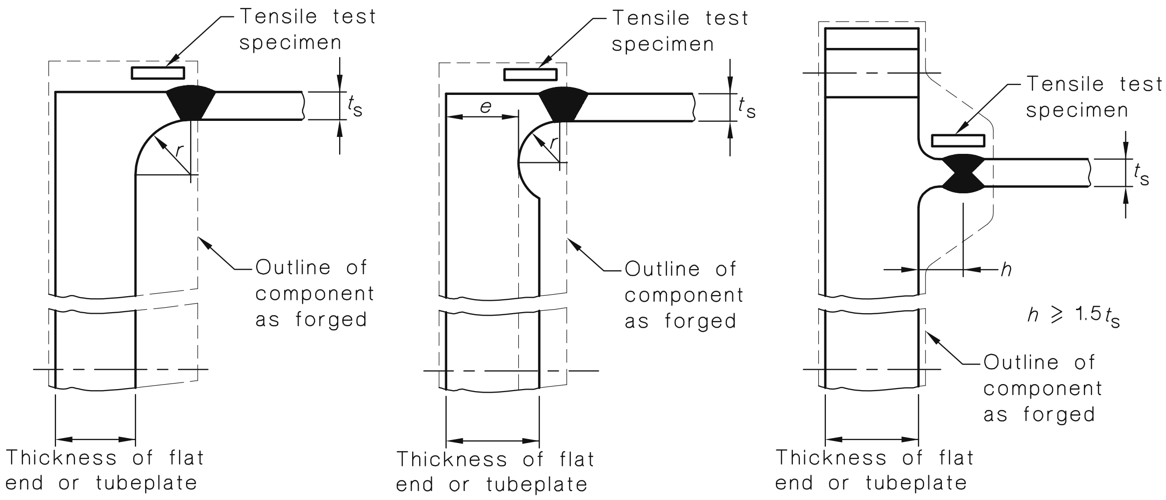
NOTE: For Notes, see end of this Figure.

FIGURE 3.17.12 (in part) ATTACHMENTS OF TUBEPLATE TO SHELL



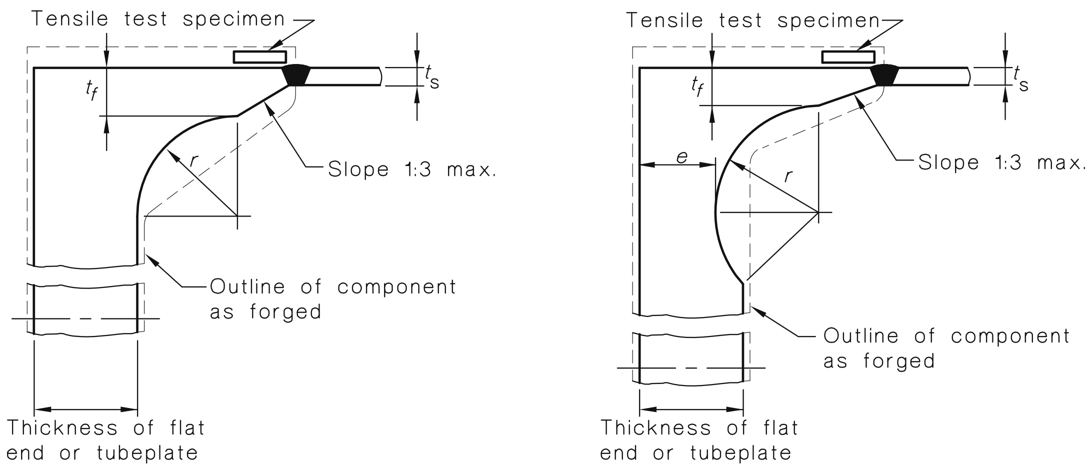
(u) (v) (w)

Clamped tubeplates (joints may be full or narrow face)



(Notes 6,7,8) (Notes 6,7,8) (Notes 6,7,8)

(x) (y) (z)



(Notes 6,7,8) (Notes 6,7,8)

(aa) (bb)

Tubeplates, supported or unsuported—forged type with hub (Note 8)

FIGURE 3.17.12 (in part) TYPICAL ATTACHMENTS OF TUBEPLATE TO SHELL

NOTES TO FIGURE 3.17.12:

1 Supported tubeplate has 80% minimum of pressure load on tubeplate carried by tubes.
2 t_s = nominal thickness of shell minus corrosion allowance.
3 For standard weld preparation see Figure 3.19.3(D).
4 Tubeplate must be checked for laminations.
5 Class 3 vessels only unless full penetration can be proven.
6 Not permissible, if machined from rolled plate.
7 The tensile test specimen may be located, when possible, inside the forged hub, instead of outside as shown.
8 Attachments shown in (x) to (bb) may be used with or without a backing ring where the weld is constructed to provide a welded joint efficiency of at least 0.9 and is inspected by magnetic particle or penetrant flaw detection methods.
9 Attachments in Group F or Group G steels shall be full penetration butt welds (see Clause 3.5.1.6).

FIGURE 3.17.12 (in part) TYPICAL ATTACHMENTS OF TUBEPLATE TO SHELL

3.18 OPENINGS AND REINFORCEMENTS

3.18.1 General

The requirements in this Clause (3.18) apply to openings and their reinforcement in cylinders, cones, spheres, and dished and flat ends. They are based on the stress intensification created by the presence of a hole in an otherwise symmetrical section, and the use of the area-replacement method, and reflect experience with vessels designed for relatively low external loads and thermal effects. The alternative area-replacement method given in AS 4041 may be used, with values of 'f' according to this Standard.

Design for external loadings shall be done according to Appendix N.

NOTE: Examples may include loads due to thermal expansion, or unsupported weight of connected piping. Such loads might become relevant in the case of unusual designs or under conditions of cyclic loading.

3.18.2 Notation

For the purpose of this Clause (3.18), the following notation applies:

t = calculated thickness of a seamless shell or end as defined in Clause 3.18.7.2, or flat end as defined in Clause 3.15 (exclusive of added allowances, see Clause 3.4.2), in millimetres

t_b = calculated thickness of a seamless nozzle wall required for pressure load plus external loads, if any (exclusive of added allowances, see Clause 3.4.2), in millimetres

T_1 = nominal thickness of the vessel wall, less corrosion allowance, in millimetres. For piping, the nominal thickness, less manufacturing undertolerance (as per the pipe specification), less corrosion allowance

T_{b1} = nominal thickness of nozzle wall, less corrosion allowance, in millimetres

T_{r1} = nominal thickness or height of reinforcing element, less corrosion allowance (see Figure 3.18.10), in millimetres

d = diameter of the finished opening in the plane under consideration plus twice the corrosion allowance (see Figure 3.18.10), in millimetres

d_m = mean diameter of the finished opening in the plane under consideration plus twice the corrosion allowance (see Figure 3.18.10), in millimetres

D = inside diameter of cylindrical or conical section or sphere, plus twice the corrosion allowance, in millimetres

c = corrosion allowance, in millimetres

h = internal depth of dishing, in millimetres

F = a factor for determining required reinforcement (see Figure 3.18.7 and Clause 3.18.7.2)

f_{r1} = design strength of set through nozzle divided by design strength of shell or end (but no more than 1.0)

= 1 for a set on nozzle

f_{r2} = design strength of nozzle wall extended beyond the shell thickness divided by design strength of shell or end (but no more than 1.0)

f_{r3} = lesser of f_{r2} and f_{r4}

f_{r4} = design strength of compensating plate divided by design strength of shell or end (but no more than 1.0)

η = 1.0 where an opening is in the plate; or

= the joint efficiency obtained from Table 3.5.1.7 where any part of the opening passes through any other welded joint

K_1 = a factor depending on the ratio $D/2h$ and defining equivalent spherical radius (see Table 3.18.7.2)

L = distance from centre of opening to the centre of an adjacent opening, in millimetres

A_1 = area in excess thickness of the vessel wall available for reinforcement, in square millimetres (see Clause 3.18.10)

A_2 = area in excess thickness of the nozzle wall available for reinforcement, in square millimetres (see Clause 3.18.10)

3.18.3 Shape of opening

Openings shall preferably be circular, but may be elliptical or obround (i.e. formed by two parallel sides with semi-circular ends).

The opening made by a pipe or a circular nozzle, the axis of which is not perpendicular to the vessel wall or end, may be considered an elliptical opening for design purposes.

When the long dimension of an elliptical or obround opening exceeds twice the short dimension, the reinforcement across the short dimension shall be increased as necessary to provide against excessive distortion due to any twisting moment.

For Class 1H and 2H vessels, the ratio of the diameter along the major axis to the diameter along the minor axis of the finished opening shall not exceed 1.5.

Openings may be of other shape than given above, provided that all corners have a suitable radius and the vessel is at least as safe as with the above openings. Where the openings are of such proportions that their strength cannot be calculated with sufficient assurance, or where doubt exists as to the safety of a vessel with such openings, the part of the vessel shall be subject to a proof hydrostatic test (see Clause 5.12).

3.18.4 Size of openings

3.18.4.1 *In cylindrical, conical and spherical shells*

Properly reinforced openings in cylindrical, conical and spherical shells need not be limited as to size. The requirements in Clause 3.18 for the reinforcement of openings shall apply within the following opening sizes:

(a) For vessels equal to or less than 1500 mm inside diameter: one-half of vessel diameter but not to exceed 500 mm.

(b) For vessels greater than 1500 mm inside diameter: one-third of vessel diameter but not to exceed 1000 mm.

(c) For Class 1H and 2H vessels: the ratio of the largest diameter of the opening to the inside diameter of the vessel at the opening location shall not exceed 0.5.

Where larger openings are required these shall be designed using a suitable alternative method listed in Clause 3.1.3. It is recommended that—

(i) the reinforcement provided be distributed close to the junction (a provision of about two-thirds of the required reinforcement within a distance of 0.25*d* on each side of the finished opening is suggested);

(ii) special consideration be given to the fabrication details used and inspection employed on critical openings;

(iii) reinforcement often may be advantageously obtained by use of heavier shellplate for a vessel course or inserted locally around the opening; and

(iv) welds may be ground to concave contour and the inside corners of the opening rounded to a generous radius to reduce stress concentrations.

The degree of inspection and non-destructive examination depends on the particular application and the severity of the intended service. Appropriate proof testing may be advisable in extreme cases of large openings approaching full vessel diameter, openings of unusual shape, and the like.

3.18.4.2 *In dished ends*

Properly reinforced openings in dished ends need not be limited as to size, but, when the opening in the end closure of a cylinder is larger than one-half of the inside diameter of the shell, it is recommended the closure be made by a reversed-curve section, or by a conical section or a cone with a knuckle radius at the large end and/or with a flare radius at the small end. The design shall comply with all the requirements of this Standard for conical sections in so far as these requirements are applicable. See Clauses 3.10 and 3.11 (see also Clause 3.18.6).

3.18.4.3 *In flat ends*

No size limits apply to openings in flat ends.

3.18.5 Location of openings

3.18.5.1 *Other than unreinforced openings in accordance with Clause 3.18.6*

Openings shall be located clear of structural discontinuities, e.g. supports, junctions between conical and cylindrical sections, by at least three times the shell or end thickness except where the design of the opening is proved adequate as required by Clause 3.1.3. (See also Clause 3.5.1.3.)

For Class 1H and 2H vessels the following shall apply:

(a) Openings are not permitted in the knuckle area unless special analysis is performed (refer to Clause 3.1.3);

(b) The distance between the centrelines of adjacent nozzles shall not be less than the sum of their internal largest opening radii. The calculated distance is—

$$\sqrt{\left(\frac{l_c}{2}\right)^2 + \left(\frac{l_1}{3}\right)^2}$$

where

l_c = the component of the centreline distance in the circumferential direction,

l_l = the component of the centreline distance in the longitudinal direction

For openings in a head, or for openings along the longitudinal axis of a cylindrical shell, $l_c = 0$.

For openings around the circumference of a cylindrical shell, $l_l = 0$.

3.18.5.2 *Orientation of non-circular openings*

Non-circular openings in cylindrical or conical shells should be arranged so the minor axis is coplanar with the longitudinal axis of the shell.

3.18.5.3 *In or adjacent to welds*

The following applies to openings in or adjacent to welds:

(a) An opening which is fully reinforced in accordance with Clause 3.18.7 may be located in a welded joint.

(b) Unreinforced openings (see Clause 3.18.6) may be located in a butt-welded joint, provided that—

  (i) the openings are machine cut and comply with the requirements for reinforcement in Clause 3.18.7 or Clause 3.18.12; or

  (ii) the welds meet Class 1 weld acceptance standards for a length equal to three times the diameter of the opening with the centre of the hole at mid-length.

  Defects that are completely removed in cutting the hole shall not be considered in judging the acceptability of the weld.

(c) Where more than two unreinforced openings (see Clause 3.18.6) are in line in a welded joint, the requirements for joint and ligament efficiency shall be met or the openings shall be reinforced in accordance with Clause 3.18.7 or 3.18.12.

(d) Unreinforced openings (see Clause 3.18.6) in solid plate shall be placed no closer to the edge of a butt-weld than 13 mm for plates 38 mm thick or less, except when the adjacent weld satisfies the requirements of (b).

(e) For Class 1H and 2H vessels, no unreinforced opening shall have its center closer than $2.5\sqrt{(R_m T_1)}$ to the edge of a locally stressed area in the shell or end, where a locally stressed area means any area in the shell or end where the primary local membrane stress exceeds $1.1f$, but excluding those areas where such primary local membrane stress is due to an unreinforced opening, and where R_m is the mean radius of the shell or end at the location of the opening.

3.18.6 Unreinforced openings

3.18.6.1 *Single openings*

Single openings on vessels not subject to rapid fluctuations in pressure are not required to have reinforcement other than that inherent in the construction, under the following conditions:

(a) The maximum finished opening, d, as defined in Clause 3.18.7.2 is—

  (i) where the shell or end nominal thickness is not greater than 10 mm........90 mm;

  (ii) where the shell or end nominal thickness is greater than 10 mm..............65 mm.

(b) Threaded, studded or expanded connections (see Figure 3.19.6(a), (b) and (c)) in which the opening in the shell or end is not greater than for 65 mm OD pipe.

(c) For openings in flat ends, the requirements in Item (a) apply. Additionally, the opening size shall be less than 0.25 × the end diameter or the shortest span.

(d) For circular openings in Class 1H, 1S, 2H and 2S, the opening diameter (d) shall not exceed $0.2\sqrt{(R_{\mathrm{m}}T_{\mathrm{l}})}$

where

R_{m} = mean radius of the shell at the opening.

3.18.6.2 *Multiple openings*

Two unreinforced openings may be located in the cylindrical, conical or spherical portion of a vessel, provided that the projected width of the ligament between any two adjacent openings shall be at least equal to the diameter of the larger opening, unless the ligament efficiency has been taken into account in the requirements of Clause 3.7. Alternatively, if the areas of all openings are wholely within a diameter equal to the maximum bore size as per Clause 3.18.6.1(a), and if the ligaments are equal to or greater than the smaller of 10 mm or the shell/end nominal thickness, then no further reinforcement is required.

When more than two unreinforced openings are provided, the ligament efficiency shall comply with the requirements of Clause 3.6.

For Class 1H, 2H, 1S and 2S vessels:

(a) Where there are two or more unreinforced openings within any circle of diameter $2.5\sqrt{(R_{\mathrm{m}}T_{\mathrm{l}})}$, then the sum of the diameters of such openings shall not exceed $0.25\sqrt{(R_{\mathrm{m}}T_{\mathrm{l}})}$.

(b) No two unreinforced openings shall have their centres closer to each other than 1.5 times the sum of their diameters where the distance is measured on the inside of the vessel wall.

3.18.7 Reinforcement of openings in shells and dished ends

3.18.7.1 *General*

The requirements of this Clause (3.18.7) apply to all openings other than—

(a) small unreinforced openings covered by Clause 3.18.6;

(b) openings in flat ends covered by Clause 3.18.9;

(c) openings designed as reducer sections covered by Clauses 3.10 and 3.11; and

(d) large end openings covered by Clause 3.18.4.2.

Reinforcement shall be provided in amount and distribution such that the area requirements for reinforcement are satisfied for all planes through the centre of the opening and normal to the vessel surface. For a circular opening in a cylindrical shell, the plane containing the axis of the shell is the plane of greatest loading due to pressure. For a single opening, not less than one-half of the required reinforcement shall be provided on each side of the centreline of the opening.

Reinforcement may be in the form of increased thickness of shell or end near the opening; increased thickness of nozzle; special fittings of increased thickness; reinforcing rings or plates around the opening on the outside and/or inside of the vessel; or by yokes of special design.

NOTE: Reinforcement located nearest the opening is most effective and stress concentration is reduced where the reinforcement is approximately equal on the inside and outside of the vessel wall.

3.18.7.2 *Reinforcement area required in shells, dished ends and cones subject to internal pressure—single openings*

The total cross-sectional area of reinforcement, in square millimetres, required in any given plane for a vessel subject to internal pressure shall be no less than the area, A, as determined in the following equation:

$$A = dtF + 2T_{b1}\, t\, F(1 - f_{r1}) \qquad \ldots 3.18.7.2$$

where

d = the finished corroded diameter of a circular opening, or finished corroded dimension of a non-circular opening (chord length at the mid-surface of the thickness, excluding excess thickness available for reinforcement) in the plane under consideration, in millimetres.

F = correction factor which compensates for the variation in pressure stresses on different planes taken through the reinforcement at varying angles to the axis of a vessel. A value of 1.0 shall be used for all configurations except that Figure 3.18.7 may be used for reinforced openings in cylindrical shells and cones where the reinforcement is integral with the nozzle

t = required thickness of a seamless shell (based on circumferential stress) or end calculated in accordance with this Standard for the calculation pressure, in millimetres, except—

(a) when the opening and its reinforcement are entirely within the spherical portion of a torispherical end, t is the thickness required by Clause 3.7 for a seamless sphere having radius equal to the crown radius of the end;

(b) when the opening is in a cone, t is the thickness required for a seamless cone of diameter, D, measured where the nozzle axis pierces the inside wall of the cone; or

(c) when the opening and its reinforcement are in an ellipsoidal end and are located entirely within a circle the centre of which coincides with the centre of the end and the diameter of which is equal to 80% of the shell diameter, t is the thickness required for a seamless sphere of radius K_1D (for K_1 see Table 3.18.7.2).

TABLE 3.18.7.2

VALUES OF SPHERICAL RADIUS FACTOR K_1

$\frac{D}{2h}$	3.0	2.8	2.6	2.4	2.2	2.0	1.8	1.6	1.4	1.2	1.0
K_1	1.36	1.27	1.18	1.08	0.99	0.90	0.81	0.73	0.65	0.57	0.50

NOTE: Values of K_1 for intermediate ratios may be obtained by linear interpolation.

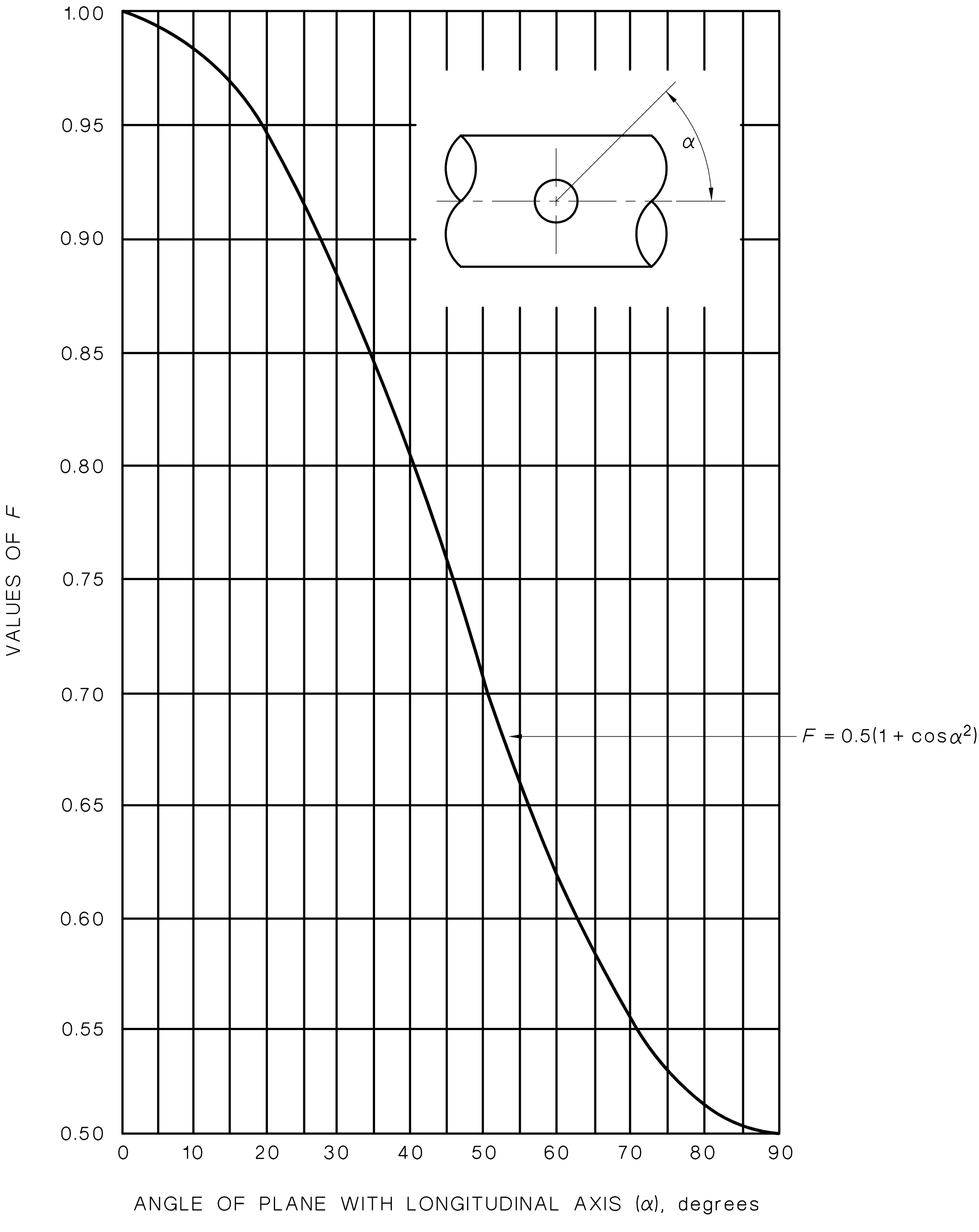


FIGURE 3.18.7 FACTOR *F* FOR REQUIRED REINFORCEMENT AREA

3.18.7.3 *Reinforcement required in shells and dished ends subject to external pressure—single openings*

A2 In shells and dished ends subject to external pressure, the following requirements apply:

(a) For openings in single-walled vessels subject to external pressure, the required reinforcement area, A, shall be not less than 50% of that required by Clause 3.18.7.2 excepting that the value of t used in the determination of area, A, in accordance with Clause 3.18.7.2, shall be the wall thickness required by this Standard for vessels subject to external pressure.

A2 (b) For multiple wall vessels subject to internal and external pressures, nozzle reinforcement area in each wall area *A*, shall each satisfy the requirements of Clauses 3.18.7.2 and 3.18.7.3(a), as appropriate; and further, when there is pressure in the space between the vessel walls only, the opening in each wall may be assumed to be stayed by the common nozzle.

3.18.7.4 *Reinforcement required in shells and dished ends subject to alternate internal and external pressures—single openings*

Reinforcement of vessels subject to alternate internal and external pressures shall meet the requirements of Clause 3.18.7.2 for internal pressure and of Clause 3.18.7.3 for external pressure applied separately.

3.18.7.5 *Reinforcement of multiple openings*

Where two adjacent openings are spaced at less than twice their average diameter so that their limits of reinforcement overlap, the two openings (or similarly a greater number of openings) shall be reinforced in accordance with Clause 3.18.7 with a combined reinforcement that has an area equal to the combined area of the reinforcement that would be required for the separate openings. No portion of the cross-section shall be considered as applying to more than one opening, or be evaluated more than once in a combined area (see Figure 3.18.7.5).

The available area of the shell or end between openings having an overlap area shall be proportioned between the openings by the ratio of their diameters. If the area between the openings is less than 50% of the total required for the 2 openings, the openings shall be subjected to special analysis (refer to Clause 3.1.3).

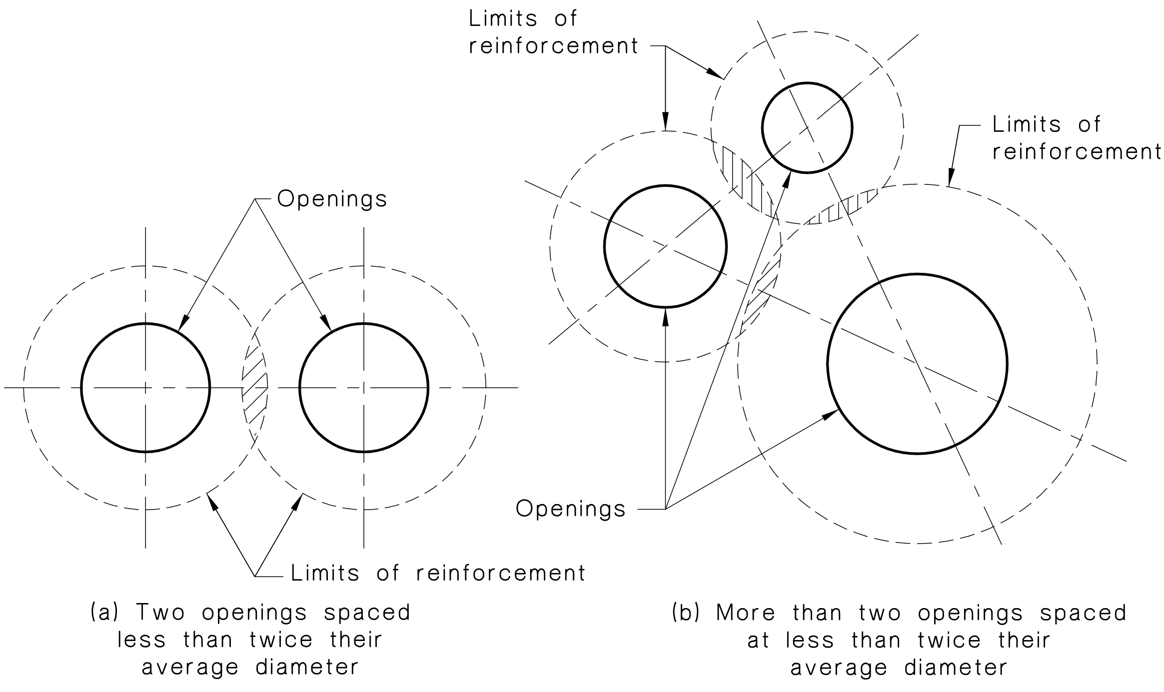


FIGURE 3.18.7.5 EXAMPLES OF MULTIPLE OPENINGS

3.18.7.6 *More than two adjacent openings*

Where more than two adjacent openings are to be provided with a combined reinforcement, the area of reinforcement between any two openings shall be at least equal to 50 percent of the total required for these two openings. The minimum distance between centres of any two of these openings should be at least 1.33 times their average diameter.

Where two adjacent openings have a distance between centres less than 1.33 times their average diameter, no credit for reinforcement shall be given for metal in the vessel wall between the openings.

3.18.7.7 *Number and arrangement unlimited*

Any number of closely spaced adjacent openings, in any arrangement, may be reinforced for an assumed opening of a diameter enclosing all such openings.

3.18.7.8 *Reinforcement by thicker section*

Where a group of openings is reinforced by a thicker section butt-welded into the shell or end, the edges of the inserted section shall be tapered as prescribed in Clause 3.5.1.8.

3.18.7.9 *Series of openings*

Where there is a series of openings arranged in a regular pattern in a cylindrical shell, the ligaments between openings shall be calculated and reinforced in accordance with Clause 3.6.

3.18.8 Flanged openings in dished ends

3.18.8.1 *Made by inward or outward flanging*

Openings in shells and dished ends made by inward or outward flanging of the plate shall meet the requirements of Clause 3.18.6 for unreinforced openings or Clause 3.18.7 for openings that require reinforcement.

3.18.8.2 *Width of bearing surface*

For internally fitted manhole doors with flat gaskets see Clause 3.15.5.1 for the width of the bearing surface.

3.18.8.3 *Shell or end thickness to be maintained*

Flanging of the opening shall not reduce the shell or end thickness below the minimum thickness required by Clauses 3.7 to 3.13, as appropriate.

3.18.8.4 *Flange thickness*

The thickness of the flange may be less than the thickness in Clause 3.18.8.3 but shall be not less than the thickness required for a cylindrical shell having a diameter equal to the major dimension of the opening.

3.18.8.5 *Flange cross-section*

The dimensions of the flange at any cross-section shall be in accordance with Figure 3.18.10(c).

3.18.9 Reinforcement required for openings in flat ends

3.18.9.1 *Application*

This Clause applies to all openings other than small openings covered by Clause 3.18.6.

3.18.9.2 *Opening less than half of end diameter or shortest span*

Flat ends that have an opening with a diameter, d, in millimetres not exceeding one-half of the end diameter or shortest span, as defined in Clause 3.15, shall have a total cross-sectional area of reinforcement, A, in square millimetres, according to Equation 3.18.9.2, where t is the minimum calculated thickness of the unpierced end.

$$A = 0.5 \times dtF + 2T_{b1}tF(1 - f_{r1}) \qquad \ldots 3.18.9.2$$

3.18.9.3 *Opening more than half of end diameter or shortest span*

Flat ends that have an opening with a diameter exceeding one-half of the end diameter or shortest span, as defined in Clause 3.15, shall be designed as a reverse flange in accordance with Clause 3.21, according to the following:

(a) The terms C and G are assigned equal to the shell inside diameter, B.

(b) M_0 for gasket seating = 0.

(c) All bolt terms = 0.

These openings in bolted covers as defined in Figure 3.15.1(k) and (l) shall be designed as a reducing flange in accordance with Clause 3.21.

3.18.9.4 *Increased thickness*

As an alternative to Clause 3.18.9.2, the thickness of flat ends may be increased to provide the necessary opening reinforcement as follows:

(a) In Equations 3.15.3(1) and 3.15.4(1) of Clause 3.15 using $K/2$ or 1.33, whichever is greater, in place of K.

(b) In Equations 3.15.3(2) and 3.15.4(2) of Clause 3.15 by doubling the quantity under the square root sign.

In the case of multiple openings, the equations in (a) and (b) shall be multiplied by the square root of one divided by the ligament efficiency, where the ligament efficiency is based on the average diameter of the two openings. The minimum ligament is limited to $0.25 \times$ the smaller diameter of the two adjacent openings.

3.18.10 Limits of available reinforcement

3.18.10.1 *Boundaries of area for reinforcement*

The boundaries of the cross-sectional area, in any plane normal to the vessel wall and passing through the centre of the opening, within which metal must be located in order to have value as reinforcement, are designated as the limits of reinforcement for that plane (see Figure 3.18.10).

3.18.10.2 *Limits of reinforcement parallel to vessel wall*

The limits of reinforcement, measured parallel to the vessel wall, shall be at a distance, on each side of the axis of the opening, equal to the greater of—

(a) the diameter of the finished opening plus twice the corrosion allowance, i.e. d on Figure 3.18.10; and

(b) the radius of the finished opening plus the corrosion allowance plus the corroded thickness of the vessel wall, plus the thickness of the nozzle wall, i.e. $(0.5d + T_1 + T_{b1})$ from Figure 3.18.10.

3.18.10.3 *Limits of reinforcement normal to vessel wall*

The limits of reinforcement, measured normal to the vessel wall, shall conform to the contour of the surface at a distance from each surface equal to the smaller of—

(a) 2.5 times the nominal shell thickness less corrosion allowance; and

(b) 2.5 times the nozzle-wall thickness less corrosion allowance, plus the thickness of added reinforcement exclusive of weld metal on the side of the shell under consideration, except that the limits from Items (a) and (b) may be exceeded provided it is not in excess of—

$$0.8(dT_{b1})^{0.5} + T_{r1} \qquad \ldots 3.18.10.3$$

For flanged-in openings in dished ends, the maximum depth that may be counted as reinforcement is $(dT_1)^{1/2}$ as shown in Figure 3.18.10(c).

If a highly-stressed area (e.g. a flange hub) falls within the calculated limits, then the limits shall be reduced appropriately.

3.18.10.4 *Reinforcing metal*

Metal within the limits of reinforcement that may be considered to have reinforcing value shall include the following:

(a) Metal in the vessel wall over and above the thickness required to resist pressure and the thickness specified as corrosion allowance. The area in the vessel wall available as reinforcement is the larger of the values of A_1 given by the equations—

$$A_1 = (\eta T_1 - Ft)d - 2T_{b1}(\eta T_1 - Ft)(1 - f_{r1}) \qquad \ldots 3.18.10.4(1)$$

$$A_1 = 2(\eta T_1 - Ft)(T_1 + T_{b1}) - 2T_{b1}(\eta T_1 - Ft)(1 - f_{r1}) \qquad \ldots 3.18.10.4(2)$$

(b) Metal over and above the thickness required to resist pressure and the thickness specified as corrosion allowance in that part of a nozzle wall extending outside the vessel wall. The maximum area in the nozzle wall available as reinforcement is the smaller of the values of A_2 given by the equations—

$$A_2 = (T_{b1} - t_b)5T_1 f_{r2} \qquad \ldots 3.18.10.4(3)$$

$$A_2 = (T_{b1} - t_b)(5T_{b1} + 2T_{r1}) f_{r2} \qquad \ldots 3.18.10.4(4)$$

except that this limit may be exceeded, provided that it is not in excess of—

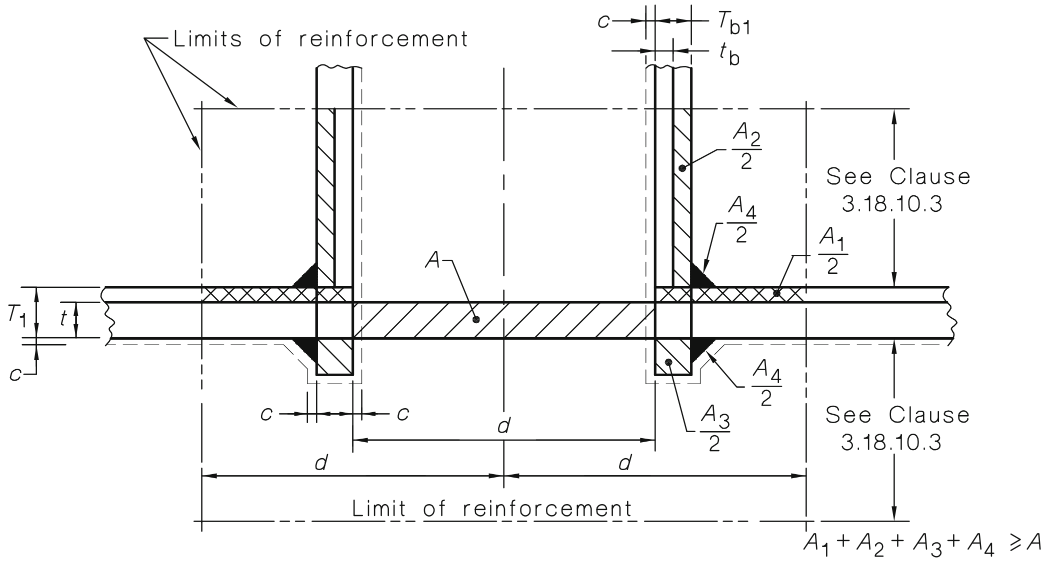
$$A_2 = (T_{b1} - t_b)\{1.6(dT_{b1})^{1/2} + 2T_{r1}\}f_{r2} \qquad \ldots 3.18.10.4(5)$$

All metal in the nozzle wall extending inside the vessel wall and within the limits of Clause 3.18.10.3 above may be included after proper deduction for corrosion allowance on all the exposed surface is made and adjusted by factor f_{r2} (see A_3 in Figure 3.18.10). No allowance shall be taken for the fact that a differential pressure on an inwardly extending nozzle may cause opposing stress in the shell around the opening.

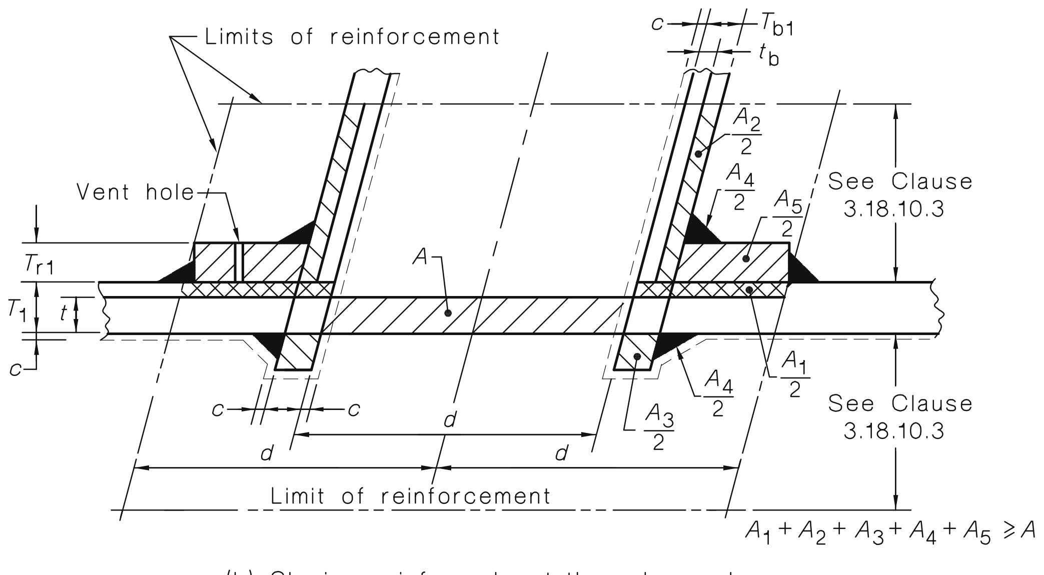
(c) Metal added as reinforcement, and metal in attachment welds (see A_5 and A_4 respectively in Figure 3.18.10).

A_4 shall be adjusted by factor f_{r3} for outside shell (or compensating plate) to nozzle weld, by a factor f_{r4} for outside compensating plate to shell weld by factor f_{r2} for inside shell to nozzle weld.

A_5 shall be adjusted by factor f_{r4}.

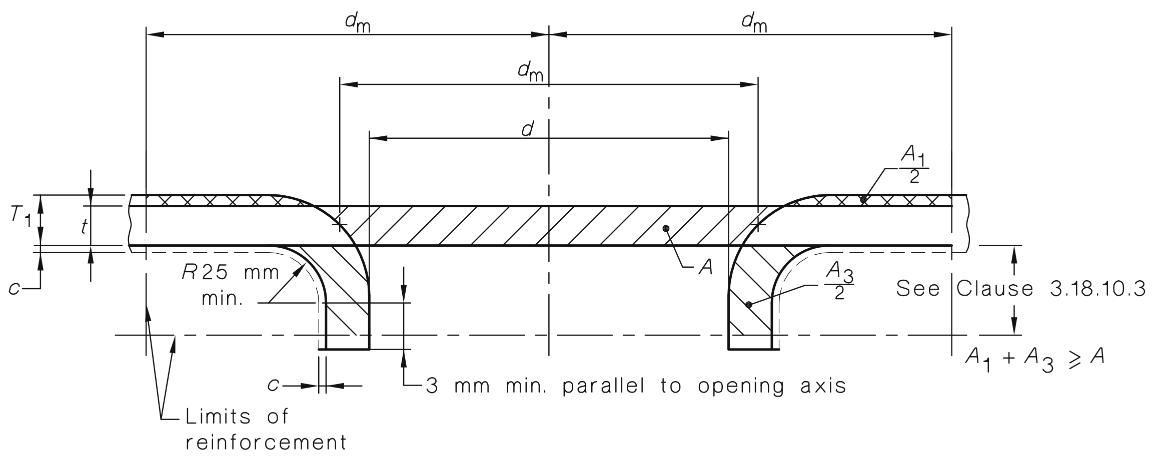


(a) Simple set-through nozzle

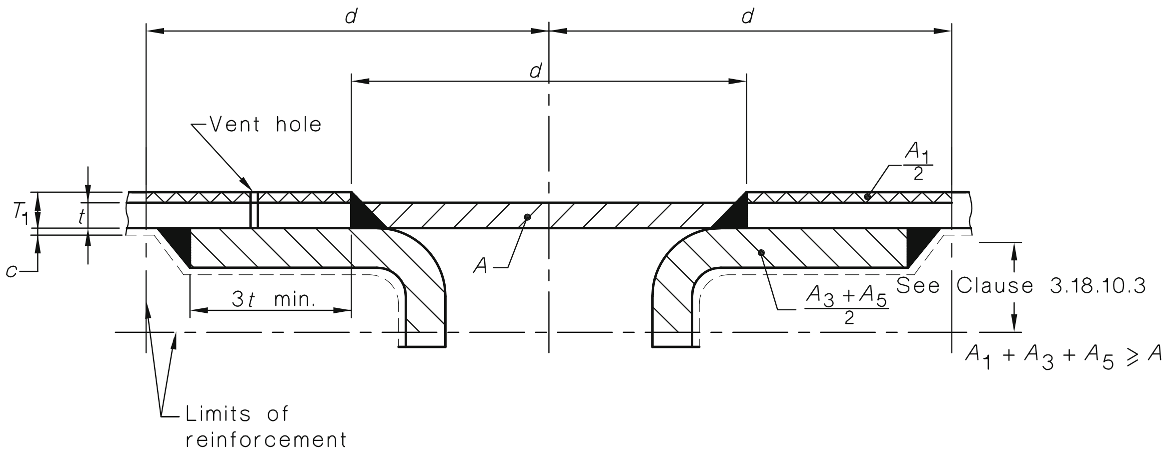


(b) Sloping reinforced set-through nozzle

FIGURE 3.18.10 (in part) REINFORCEMENT AREAS AND LIMITS FOR OPENINGS



(c) Inwardly flanged opening



(d) Inwardly flanged and welded opening

NOTE: For definition of symbols see Clause 3.18.2

FIGURE 3.18.10 (in part) REINFORCEMENT AREAS AND LIMITS FOR OPENINGS

3.18.11 Strength of reinforcement

3.18.11.1 *General*

Material used for reinforcement shall have a design strength equal to or greater than that of the material in the vessel wall, except that when such material is not available, lower strength material may be used. The strength reduction factors f_{r1}, f_{r2}, f_{r3} and f_{r4} are to take account of different strength materials, but in no case are these to exceed 1.0.

All pressure parts at openings and nozzles in vessels constructed of Group F or Group G steels shall be made of material whose specified minimum tensile strength is equal to or greater than that of the shell material, except that pipe flanges, pipe or communicating chambers may be of carbon steel, low alloy steel or high alloy steel welded to nozzle necks of the required material provided that—

(a) the joint is a circumferential butt weld located no less than $(rt)^{½}$ measured from the limit of reinforcement as defined in Clause 3.18.10 where r is the inside radius of the nozzle neck, and t is the thickness of the nozzle at the joint;

(b) the design of the nozzle neck at the joint is made on the basis of the allowable stress value of the weaker material;

(c) the slope of the nozzle neck does not exceed 3:1 for at least a distance of 1.5t from the centre of the joint; and

(d) the diameter of the nozzle neck does not exceed the limits given in Clause 3.18.4.1.

3.18.11.2 *Requirements*

Strength of attachments and detailed requirements for welded and brazed reinforcement are given in Clause 3.19.

3.19 CONNECTIONS AND NOZZLES

3.19.1 General

Pipes, nozzles and fittings shall be connected to the vessel shells and ends in accordance with this Clause (3.19). For bolted flanged connections, see Clause 3.21.

A1 In addition to the requirements of this Clause, consideration shall be given to connections subject to loadings as specified in Clause 3.2.3. The design for local nozzle and structural non-pressure loads shall be according to Appendix N.

3.19.2 Strength of attachment

A1 The following meet the requirements of Clause 3.19.2 for pressure loading and do not require strength calculations:

- Welded connections that comply with Figure 3.19.3(A), illustrations (a), (b), (c), (d), (e), (f), (g), (h), (j) and (k); Figure 3.19.3(B), illustrations (f), (g), (h), (j) and (k); Figure 3.19.3(C), illustrations (a) and (c); Figure 3.19.4, illustrations (e), (f), (k), (o) and (p); Figure 3.19.6, illustrations (g), (h) and (j); Figure 3.19.9(b), (c), (d), (e) and (f).

  NOTE: Strength calculations may be required for connections subject to loadings as specified in Clause 3.2.3 or where directed in the notes to the relevant figure.

- If A_1 is found to be greater than A in Clause 3.18 above.
- If the conditions stated in Clause 3.18.6.1 are met.

Refer to Clause 3.19.3.2 for minimum sizes of welds at attachments.

Single fillet-welded connections should be avoided where cyclic stresses are likely to occur, where thermal gradients may overstress the attachment welds, and where corrosive conditions may occur.

The strength of attachment of connections shall meet the following requirements:

(a) On each side of the plane defined in Clause 3.18.10.1, the strength of the attachment joining the vessel wall and reinforcement or parts of the attached reinforcement shall be greater than or equal to the smaller of—

  (i) the strength in tension of the cross-section of the element of reinforcement being considered; and

  (ii) the strength in tension of the area defined in Clause 3.18.7 less the strength in tension of the reinforcing area which is integral in the vessel wall as permitted by Clause 3.18.10.4(a).

(b) The strength of the attachment joint shall be considered for its entire length on each side of the plane of the area of reinforcement defined in Clause 3.18.10. For obround openings, consideration shall also be given to the strength of the attachment joint on one side of the plane transverse to the parallel sides of the opening which passes through the centre of the semi-circular end of the opening.

(c) Figure 3.19.2 shows typical paths of failure (shown as dashed lines) for set-in and set-on nozzle connections.

To check for the adequacy of strength of welds, it is required that strength of welds and attachments (force carrying capacity) is equal to, or greater than, the strength in tension of the elements. The method is shown below:

(i) *Strength of welds and attached elements* There shall be sufficient strength in the welds and attached elements. The strength of the welds and attached element shall be calculated as follows:

NOTE: Refer to Clause 3.19.3.5 for allowable stresses of welds.

Figure 3.19.2 (a), (b):

$$W_1 = \pi/2\ d_o\ t_{c41}\ f_{weld\ 41} \quad \ldots 3.19.2(1)$$

$$W_2 = \pi/2\ d_m\ t_{b1}\ f_b \quad \ldots 3.19.2(2)$$

$$W_3 = \pi/2\ d_o\ T_{sg}\ f_s \quad \ldots 3.19.2(3)$$

$$W_4 = \pi/2\ d_r\ T_{c42}\ f_{weld\ 42} \quad \ldots 3.19.2(4)$$

$$W_5 = \pi/2\ d_o\ T_{c43}\ f_{weld\ 43} \quad \ldots 3.19.2(5)$$

$$W_6 = \pi/2\ d_o\ T_{rg}\ f_r \quad \ldots 3.19.2(6)$$

where

d_o = outer diameter of the nozzle, in millimetres

d_m = mean of inside and outside diameter of the nozzle, in millimetres

d_r = outside diameter of reinforcing element, in millimetres

f_s = design stress of the shell in tension, in megapascals

f_r = design strength of reinforcement ring on plate (see Clause 3.18.11 and Table B1), in megapascals

$f_{weld\ 41}$ = design strength of weld 41 in shear, in megapascals

t_{c41} = throat thickness of weld 41, in millimetres

T_{rg} = depth of reinforcing plate to nozzle butt weld, in millimetres

T_{sg} = depth of shell to nozzle butt weld, in millimetres

W_1 = strength of nozzle outside fillet weld in shear, in newtons

W_2 = strength of nozzle wall in shear, in newtons

W_3 = strength of shell to nozzle butt weld in tension, in newtons

W_4 = strength of reinforcing plate outer fillet weld in shear, in newtons

W_5 = strength of nozzle inside fillet weld in shear, in newtons

W_6 = strength of reinforcing plate to nozzle butt weld in tension, in newtons

NOTE: Refer to Clause 3.18.2 for all other notations.

(ii) *Strength in tension of elements* The strength in tension of the element or elements of reinforcement being considered are defined as W_{11}, W_{22}, W_{33} and shall be calculated as follows:

Figure 3.19.2 (a):

$$W = (A - A_1 + 2\ T_{b1}\ f_{r1}\ (\eta\ T_1 - F_t))\ f \quad \ldots 3.19.2(7)$$

$$W_{11} = (A_2 + A_3 + A_{41} + A_{43} + 2\ T_{ba}\ T_1\ f_{p1})\ f \quad \ldots 3.19.2(8)$$

$$W_{22} = (A_2 + A_5 + A_{41} + A_{42})\, f \qquad \ldots 3.19.2(9)$$

$$W_{33} \quad (A_2 + A_3 + A_5 + A_{41} + A_{42} + A_{43} + 2\, T_{b1}\, T_1\, f_{r1})\, f \qquad \ldots 3.19.2(10)$$

Figure 3.19.2 (b)

$$W = (A - A_1)\, f \qquad \ldots 3.19.2(11)$$

$$W_{11} = (A_2 + A_{41})\, f \qquad \ldots 3.19.2(12)$$

$$W_{22} = (A_2 + A_5 + A_{41} + A_{42} + A_{43})\, f \qquad \ldots 3.19.2(13)$$

where

W = total weld load, in newtons

W_{11} = weld load for failure path 11, in newtons

f = design strength of parent metal, in megapascals

Refer to Clause 3.18.2 for all other notations.

(iii) *Check for adequate strength of welds* The strength of welds and attachments is required to be equal to, or greater than, the strength in tension of the elements as follows:

Figure 3.19.2 (a)

$$W_1 + W_3 + W_5 + W_6 \geq \text{lesser of } W \text{ and } W_{11} \qquad \ldots 3.19.2(14)$$

$$W_4 + W_2 + W_6 \geq \text{lesser of } W \text{ and } W_{22} \qquad \ldots 3.19.2(15)$$

$$W_3 + W_4 + W_5 \geq \text{lesser of } W \text{ and } W_{33} \qquad \ldots 3.19.2(16)$$

$$W_4 \geq \text{actual } A_5 f_r \qquad \ldots 3.19.2(17)$$

Figure 3.19.2(b)

$$W_1 + W_3 + W_6 \geq \text{lesser of } W \text{ and } W_{11} \qquad \ldots 3.19.2(18)$$

$$W_2 + W_4 + W_6 \geq \text{lesser of } W \text{ and } W_{22} \qquad \ldots 3.19.2(19)$$

$$W_4 \geq \text{actual } A_5\, f_r \qquad \ldots 3.19.2(20)$$

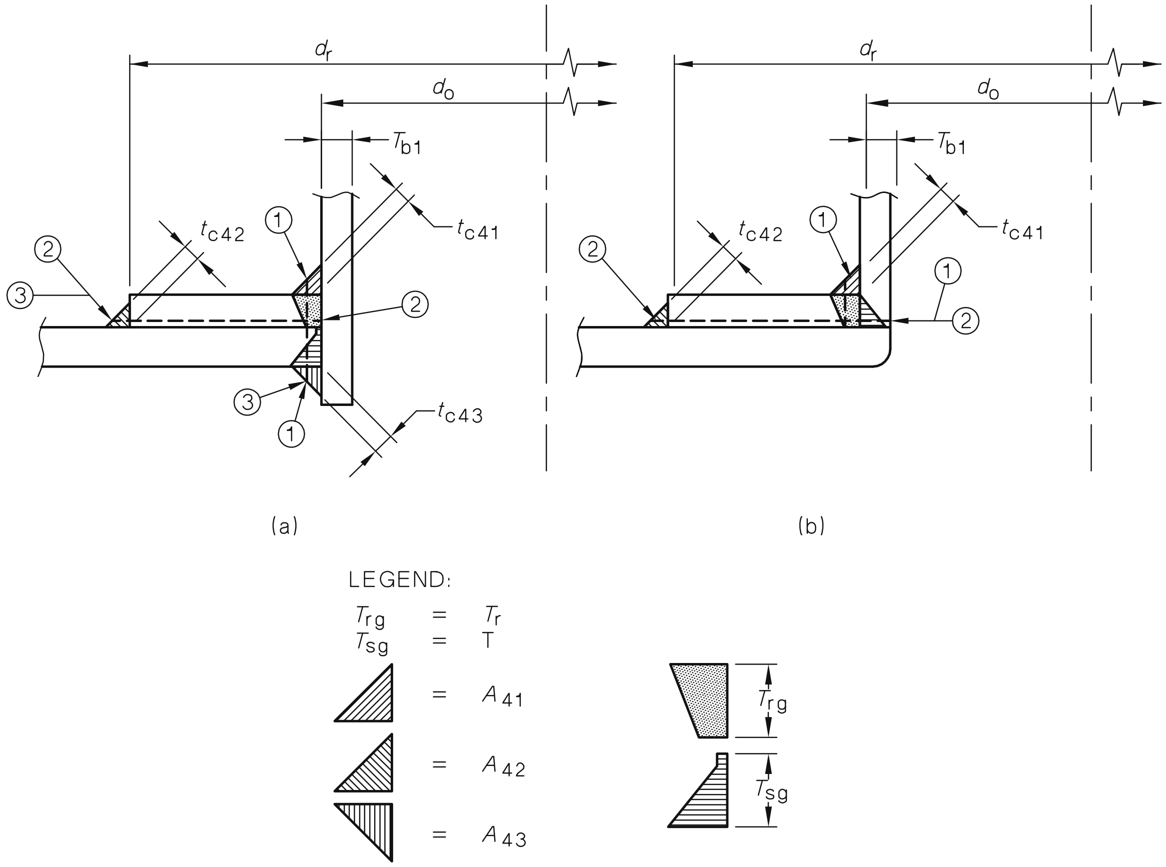


FIGURE 3.19.2 TYPICAL PATHS OF FAILURE OF WELDED CONNECTIONS

3.19.3 Welded nozzle connections and reinforcement

3.19.3.1 *Application*

Arc or gas-welded connections may be used to attach nozzles, pads and reinforcement rings or plates to weldable materials.

Where thermal gradients are high, welds with full penetration should be used, and reinforcement plates and similar construction avoided.

3.19.3.2 *Methods of attachment*

The attachment of nozzles and reinforcement shall comply with the requirements of this Clause (3.19.3.2). Figures 3.19.3(A), (B) and (C) and Figure 3.19.4 provide some acceptable details of attachment for nozzles and reinforcement. In addition, strength calculations where required by Clause 3.19.2 shall be performed.

The overall length of any obround, set-in pad attached to the shell by double fillet welds shall not exceed one-half the inside diameter of the vessel.

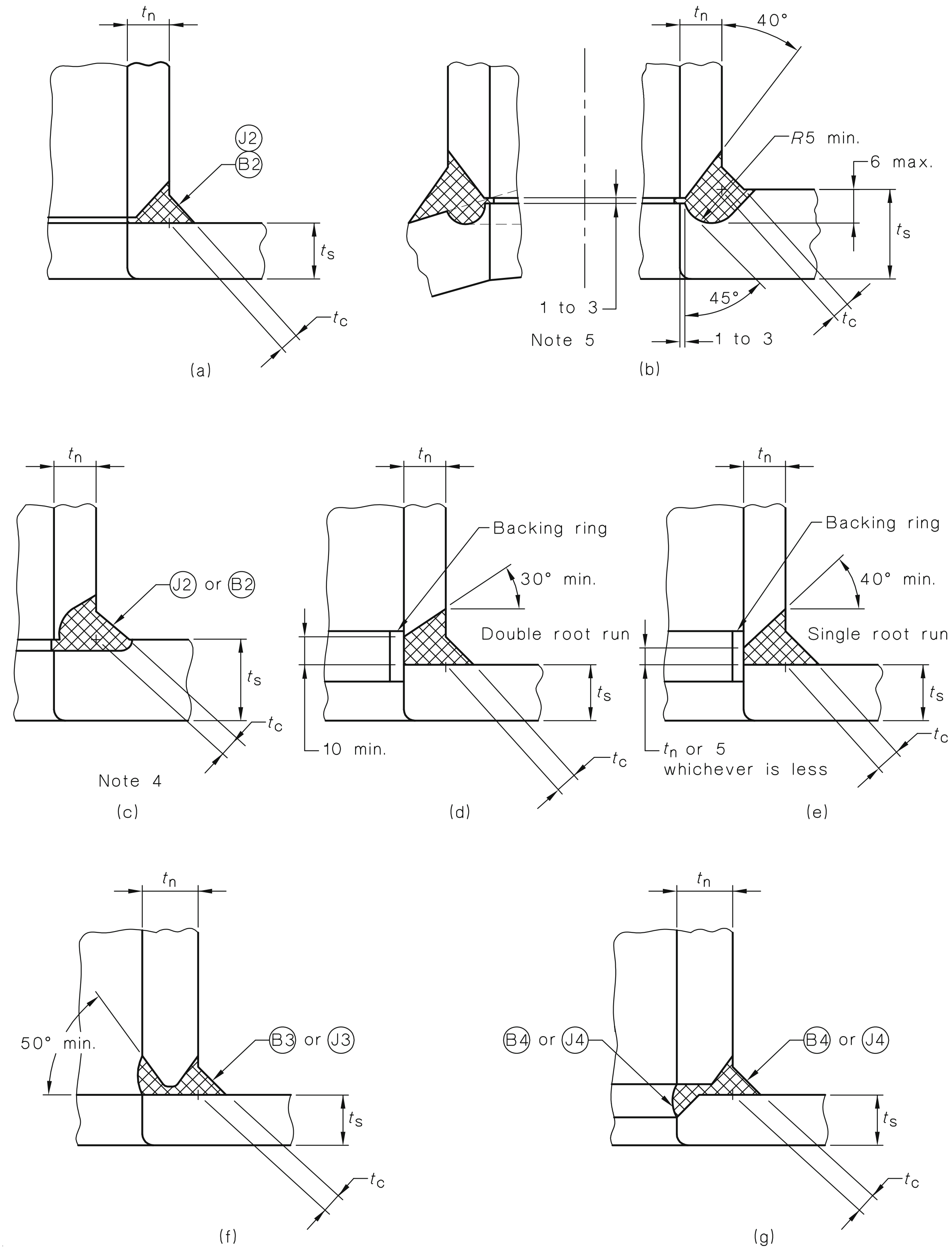
Connections involving Group F steel shall be attached with full penetration butt welds and the radius at nozzle entry equal to $t/4$ or 20 mm, whichever is the lesser.

Connections involving Group G steel shall be full penetration butt welds in accordance with Figure 3.19.9 excluding Figures (a) and (b).

3.19.3.3 *Hole for inserted connections*

The diameter of the hole cut through a shell or end for an inserted pad or nozzle shall not exceed the diameter of the pad or nozzle by more than 6 mm. The pad or nozzle shall be centrally located in the opening before welding.

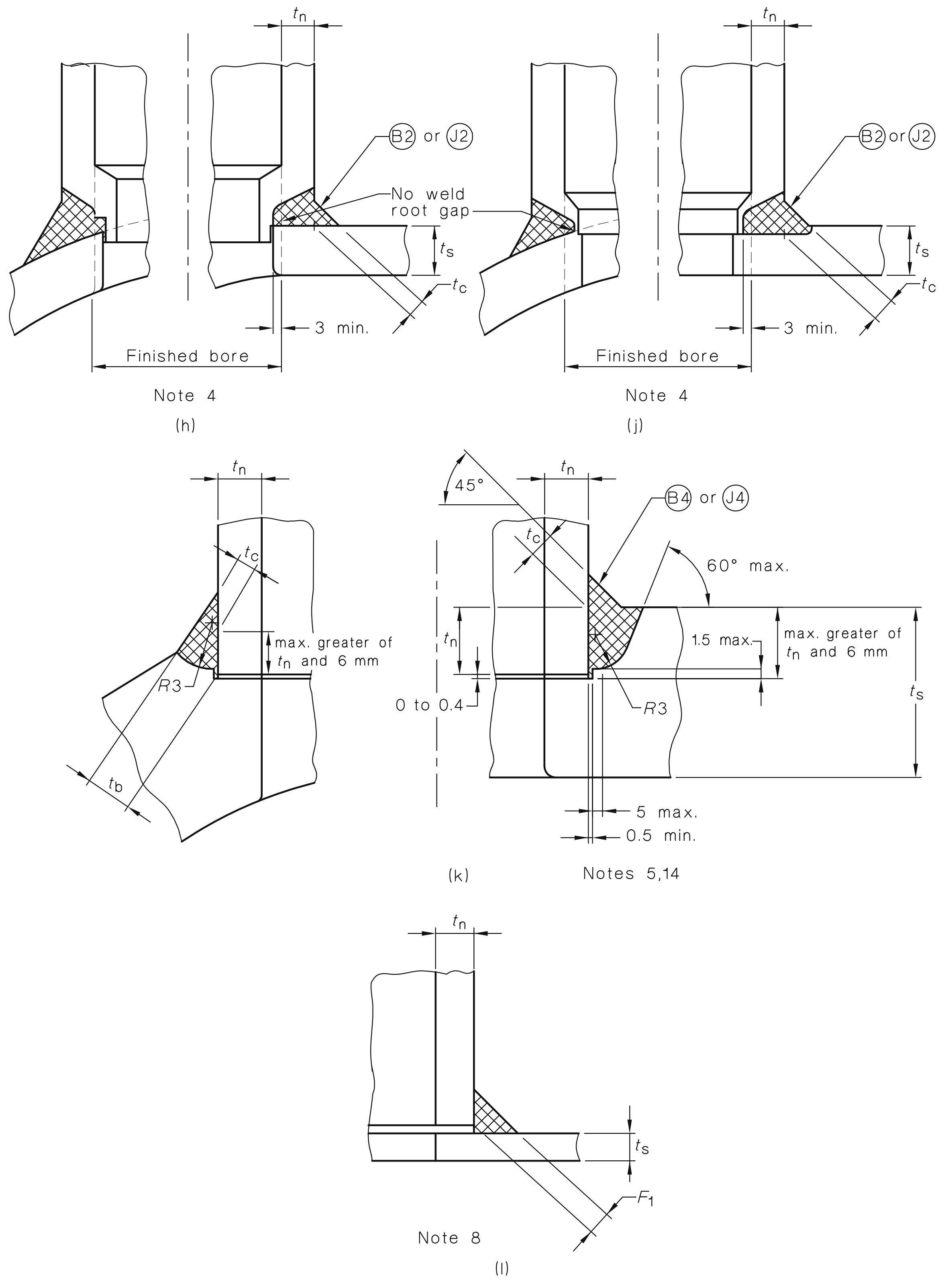
Where the surface of the hole will not be fused during welding, the hole shall not be punched and the surface shall have a smooth finish free from sharp crevices.



LEGEND and NOTES: See end of Figure 3.19.3(C).

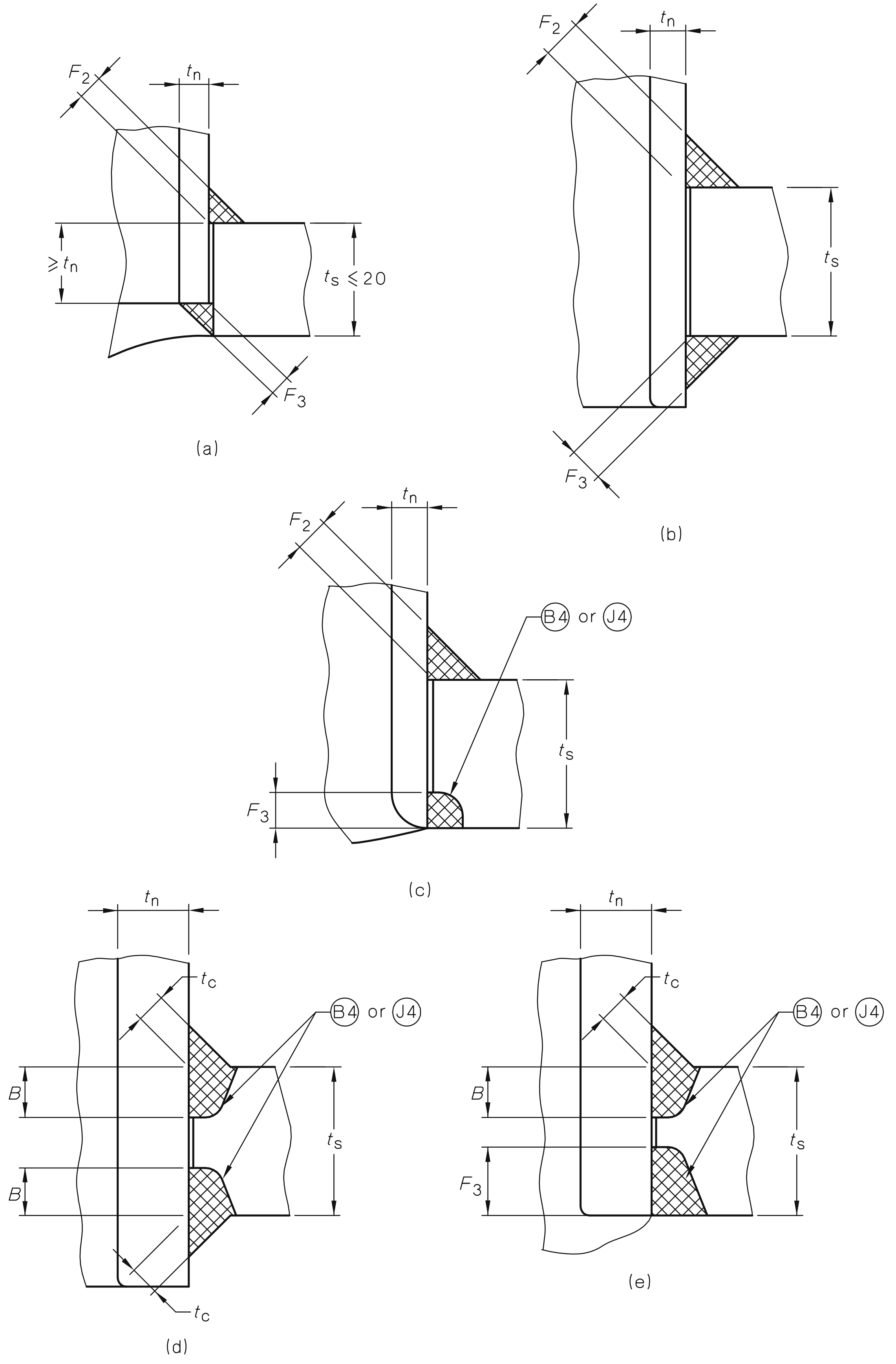
DIMENSIONS IN MILLIMETRES

FIGURE 3.19.3(A) (in part) SOME ACCEPTABLE NOZZLE CONNECTIONS—SET-ON TYPE



DIMENSIONS IN MILLIMETRES

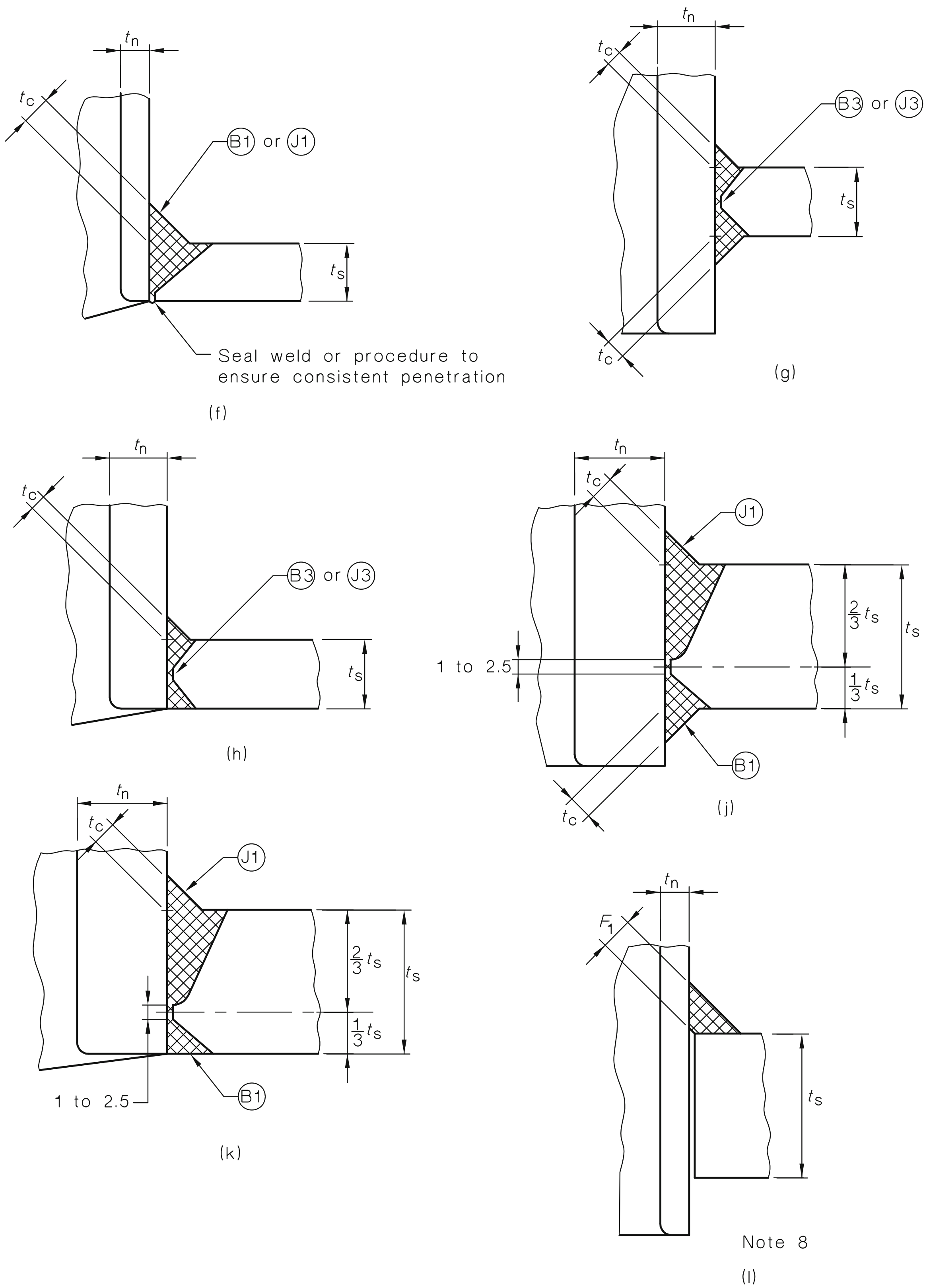
FIGURE 3.19.3(A) (in part) SOME ACCEPTABLE NOZZLE CONNECTIONS—SET-ON TYPE



LEGEND and NOTES: See end of Figure 3.19.3(C).

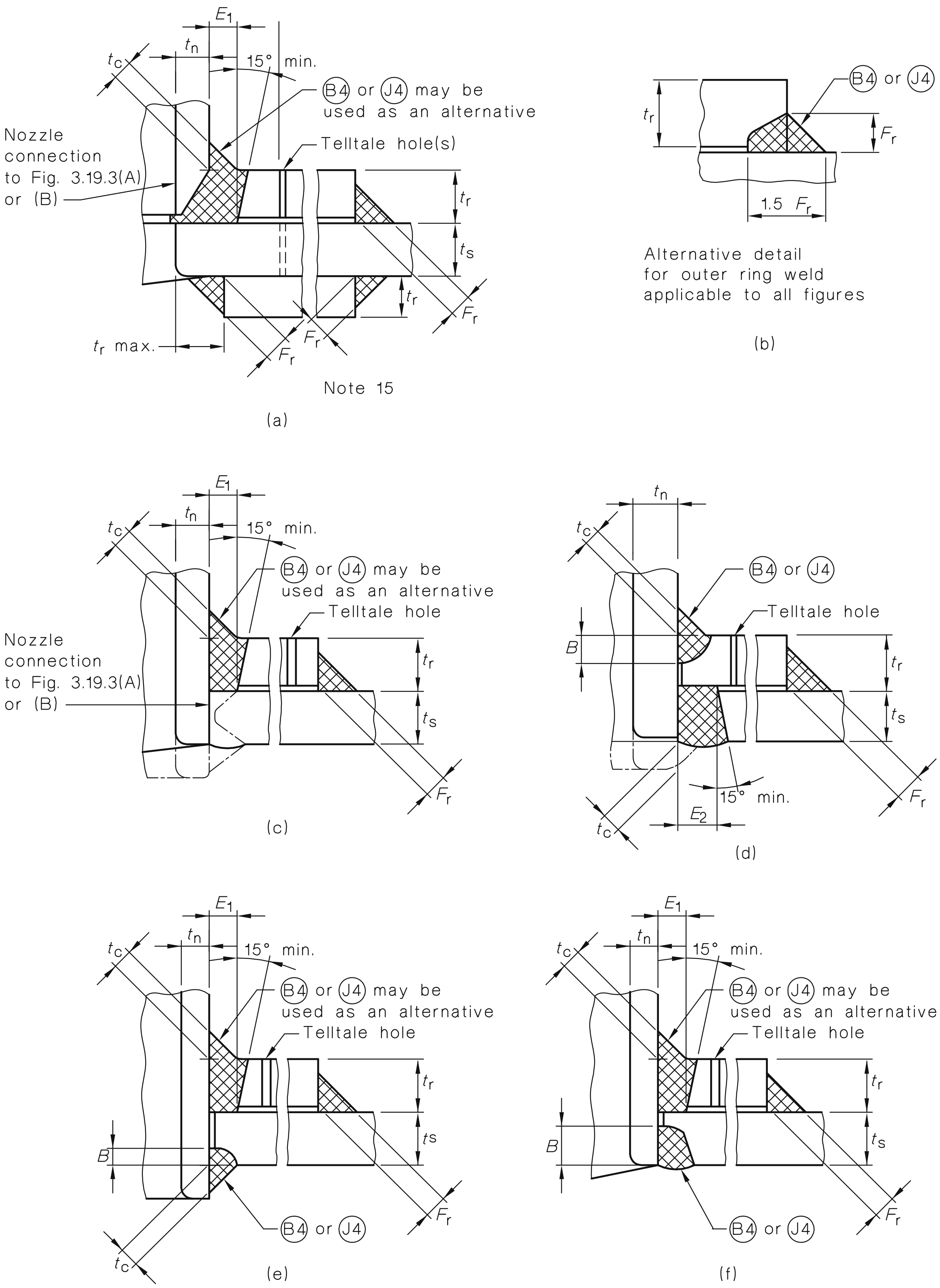
DIMENSIONS IN MILLIMETRES

FIGURE 3.19.3(B) (in part) SOME ACCEPTABLE NOZZLE CONNECTIONS—SET-IN TYPE



LEGEND AND NOTES: See end of Figure 3.19.3(C).

FIGURE 3.19.3(B) (in part) SOME ACCEPTABLE NOZZLE CONNECTIONS—SET-IN TYPE



LEGEND and NOTES: See end of Figure 3.19.3(C).

DIMENSIONS IN MILLIMETRES

FIGURE 3.19.3(C) SOME ACCEPTABLE NOZZLE CONNECTIONS WITH REINFORCING PLATES

LEGEND TO FIGURES 3.19.3(A), 3.19.3(B), and 3.19.3(C):

t_s	=	nominal thickness of vessel wall minus corrosion allowance, in millimetres
t_n	=	nominal minimum thickness of nozzle wall minus corrosion allowance, (Note 16), in millimetres
t_c	≥	$0.7t_s$, $0.7t_n$ or 6 mm, whichever is least
t_{ri}	=	nominal thickness or height of reinforcing element (outside or inside vessel) minus corrosion allowance, in millimetres
B	≥	$0.7t_s$, $0.7t_n$, or 14 mm, whichever is less
E_1	≥	$t_r/2$ or 10 mm, whichever is less
E_2	≥	$t_s/2$ or 10 mm, whichever is less
F_1	≥	1.25 t_s or 1.25 t_n whichever is less
F_2 or F_3	≥	0.7 t_s, 0.7 t_n or 6 mm, whichever is least
$F_2 + F_3$	≥	*1.25* t_s *or 1.25* t_n, whichever is least
F_4	=	0.5 t_s, 0.5 t_n or 10 mm, whichever is least
F_5	=	0.7 t_s, 0.7 t_n or 14 mm, whichever is least
F_r	≥	$0.5t_r$, 0.5 t_s or 10 mm, whichever is least

NOTES TO FIGURES 3.19.3(A), 3.19.3(B) AND 3.19.3(C):

1 See Figure 3.19.3(D) for standard nozzle weld details. Recommended departures in weld preparation angle from these details are shown on applicable sketches.

2 Backing rings must fit closely and shall be removed after welding except where otherwise agreed between the parties concerned.

3 Not allocated.

4 Connections (c), (h), (j) of Figure 3.19.3(A) are generally used for small nozzle to shell diameter ratios.

5 Connections (b) and (k) of Figure 3.19.3(A) are suitable for thicker shells but (k) is not suitable for corrosive conditions.

6 In all set-on nozzle connections the shell plate around the hole shall be examined visually for laminations prior to welding and, where practicable, for lamellar type tearing after welding.

7 Provide 3 mm minimum radius or chamfer on corner at bore or nozzle except where otherwise noted.

8 Connections (1) of Figure 3.19.3(A) and (1) of Figure 3.19.3(B) are limited to Class 3 vessels with opening in shell or end not greater than maximum permitted unreinforced opening (see also Clause 3.19.5) and are not suitable for corrosive conditions.

9 The use of partial penetration welds and connections in Figures 3.19.3(C) should be avoided where cyclic stresses are likely to occur, where thermal gradients may overstress attachments welds, where the nozzle or shell thickness exceeds 50 mm, or where high strength materials are used, or where temperatures exceed 250°C.

10 For all set on nozzle conditions, and set-in partial penetration connections, see Clause 3.19.3.3 and AS 4458 for edge finish of unwelded openings.

11 Where weld preparations (B4) or (J4) of Figure 3.19.3(D) are used for the attachment of reinforcing plates in Figure (a), (c), (e) and (f) of Figure 3.19.3(C), paths of failure shall be checked for strength.

12 Corrosion allowance shall be added to weld throat thickness, where applicable.

13 The figures show a 45° fillet weld as typical. This should not exceed 50° on the weld toe at the thinner member in Group D to J steels or where fatigue, shock, brittle fracture or exceptional external loads are important considerations.

14 Connection (k) of Figure 3.19.3(A) is permitted for nozzles up to and including 150 mm nominal bore and t_n up to and including 6 mm. This connection should not be used for corrosive conditions.

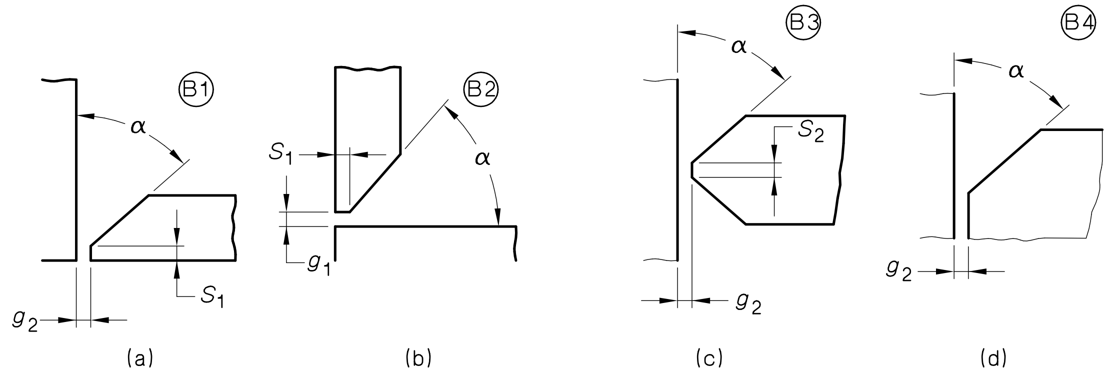
15 The connections as detailed in Figure 3.19.3(C) may have the reinforcing plate on either or both sides of the vessel wall, with corrosion allowance added to reinforcing plates on the inside wall and tell-tale holes venting to atmosphere. Where a reinforcing ring is internal, a telltale hole shall be located in the shell wall.

16 The minimum thickness of a nozzle wall (t_n) is the thinnest part of the wall of the nozzle. See Figure 3.19.9(e) for nozzles of varying wall thickness.

17 In Figure 3.19.3(A), connection (k) shall not be used for Class 1H, 2H, 1S or 2S.

18 In Figure 3.19.3(B), connections (a), (b), (c), (d) and (e) shall not be used for Class 1H, 2H, 1S or 2S.

19 In Figure 3.19.3(C), connections (d), (e) and (f) shall not be used for Class 1H, 2H, 1S or 2S.



LEGEND

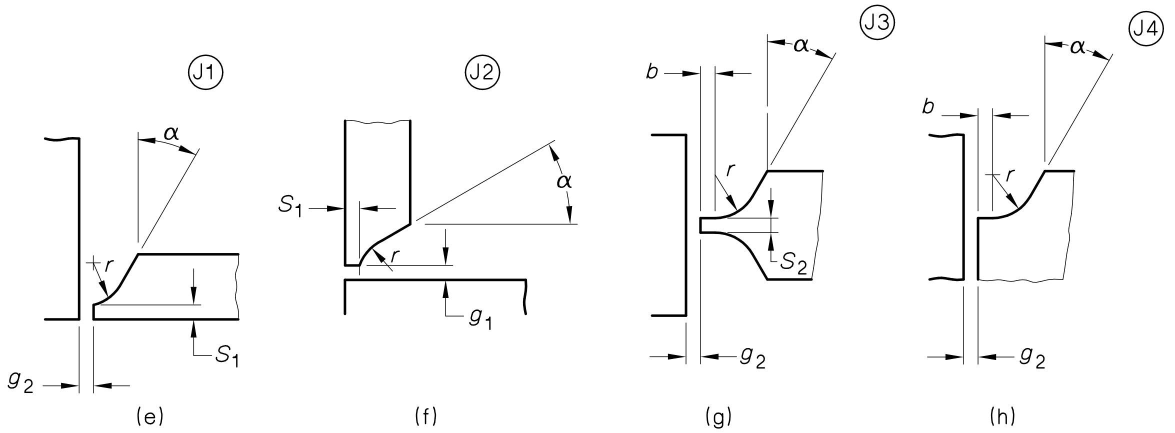
α = 50° min.

S_1 = 1.5 to 2.5 mm

S_2 = 0 to 3 mm

g_1 = 1.5 to 2.5 mm when t_b < 10 mm
1.5 to 4.0 mm when $t_b \geqslant$ 10 mm

g_2 = See Note 2



LEGEND

α = 15° to 35°

S_1 = 1.5 to 3.0 mm

S_2 = 1.5 to 3.0 mm

g_1 = 1.5 to 3.0 mm

g_2 = See Note 2

b = 0 to 3.0 mm

r = 6 to 12 mm

NOTES:

1 Weld preparations shown are regarded as prequalified preparations. Other preparations may be used, provided the manufacturer demonstrates (using appropriate tests) that welds of the required size and quality can be obtained. Discretion must be used in applying the maximum and minimum dimensions quoted which are subject to variation according to the welding procedure employed (e.g. size and type of electrodes) and also the position in which the welding is carried out.

2 It is recommended that in no case should the gap between nozzle and shell exceed 3 mm. Wider gaps increase the tendency to spontaneous cracking during welding, particularly as the thickness of the parts joined increases. For gas tungsten arc welding (GTAW) the gap is to be further reduced.

3 The use of minimum angle should be associated with the maximum radius or gap. Conversely the minimum radius or gap should be associated with the maximum angle.

FIGURE 3.19.3(D) STANDARD WELD DETAILS FOR NOZZLE CONNECTIONS

3.19.3.4 *Tell-tale holes*

Reinforcement plates which might have chambers sealed by welding shall have at least one tell-tale hole per chamber that may be tapped to 15 mm major diameter (maximum), for testing of the welding sealed off by the reinforcing plate, and for venting during any heat treatment. For flanged joints or similar applications, the end of the hole shall be clear of contact surfaces. After testing, the hole shall be exposed to atmosphere to prevent pressure build-up in the event of leakage through the attachment welds, however the hole may be filled with a material that will exclude moisture.

NOTES:

1 Tell-tale holes can admit moisture between the plate and vessel, forming a corrosion pocket. Placement of the tell-tale hole at a low point will allow more effective drainage of any moisture. In some applications, such as underground vessels, inclusion of poorly-drained tell-tale holes can be counter-productive.

2 This Clause does not refer to other reinforcement plates such as those used under attachments. Reference should be made to Clauses 3.24.7 and 3.25.3.

3.19.3.5 *Strength of welded connections*

(See Note to Clause 3.19.2.) Nozzles, other connections and their reinforcements attached to pressure parts shall have sufficient welding on either side of a line, through the centre of the opening, parallel to the longitudinal axis of the shell, to develop the strength of the reinforcing parts as required by Clause 3.19.2, through shear or tension in the weld whichever is applicable.

The strength of a weld shall be based on the nominal throat of the weld times the length of the weld measured on its inner periphery times the maximum allowable stress for the weld. The allowable stress in the weld and in the component, which may be included in the possible path of failure, shall be the following percentage of the design tensile strength for the appropriate material (see Table B1):

(a) Fillet weld in end shear equals 70 percent.

(b) Butt weld in tension equals 74 percent.

(c) Butt weld in shear equals 60 percent.

(d) Components (e.g. nozzle wall) in shear equals 70 percent.

Where the load on a weld varies from side to end shear or shear to tension, the lower of the above values shall be used.

3.19.4 Screwed and socket welded connections

3.19.4.1 *General*

Screw-threaded joints other than for screwed flanges may be used for the connection of pipes and fittings to pressure vessels within the limits set out in the following clauses. Such joints are not recommended where the connection is to be broken and remade frequently or is subject to continuous vibration, or with toxic or flammable contents. Some acceptable types are shown in Figure 3.19.4.

Screwed and expanded connections shall not be used for vessels with lethal contents (see Clause 1.7.1).

3.19.4.2 *Pipe threads*

Threads shall be in accordance with AS 1722.1, AS 1722.2, ANSI B1.20.1 or API Std 5B, or equivalent Standard.

Threads shall be right hand and may be taper-to-taper, parallel-to-parallel or taper-to-parallel.

Threads shall be true to their full length and depth and shall comply with any gauging requirements of the relevant specification.

3.19.4.3 *Size limitation—Threaded joints*

For Classes 1, 2 and 3 threaded joints other than for screwed flanges shall not exceed 65 mm OD (i.e. DN 50), except that threaded joints up to and including 90 mm OD (i.e. DN 80) may be used provided that—

(a) a taper-to-taper joint is used;

(b) the depth of engagement is not less than 25 mm; and

(c) any socket is a heavy wall type with gauged threads.

For Classes 1H, 2H, 1S and 2S, threaded joints shall not exceed 65 mm OD (i.e. DN 50).

For threaded openings in forged ends see AS 4458.

3.19.4.4 *Size limitation—Socket welded connections*

For all Classes, socket welded joints shall not exceed 65 mm OD (i.e. DN 50).

3.19.4.5 *Temperature and pressure limits*

Where temperature fluctuations could cause relaxation of the joint, threaded joints other than for screwed flanges shall be limited to a maximum metal temperature of 260°C.

Threaded joints using steel heavy pipe with ordinary sockets, both of which conform to the dimensions of AS 1074, shall be limited to the maximum pressure given in Table 3.19.4.

TABLE 3.19.4

MAXIMUM ALLOWABLE PRESSURE FOR THREADED JOINTS OTHER THAN SCREWED FLANGES

Outside diameter of pipe mm	Maximum allowable pressure, MPa		
	Taper-to-parallel joint	Parallel-to-parallel joint	Taper-to-taper joint
≤35	1.2	1.2	2.1
>35 ≤50	1.05	1.05	1.75
>50 ≤65	0.86	0.86	1.55
>65 ≤90	—	—	1.55

NOTE: AS 1074 materials are not acceptable for integral pressure parts of vessels.

Notwithstanding the above requirements, threaded joints of taper-to-taper in accordance with API 5B or ANSI B1.20.1, or parallel-to-parallel type made with components which comply with BS 3799, BS 5154 or EN 15761 (or other equivalent Standards) may be used for temperatures and pressures up to the maximum permitted by that Standard for the component having the lower pressure/temperature rating.

3.19.4.6 *Sealing*

Where threaded joints are likely to seize or corrode, sealing shall be arranged to prevent threads from coming in contact with the contained fluid. Where a sealing gasket is used, it shall be fitted so that it cannot cause inadvertent blockage of a passageway.

To facilitate tightening during assembly and to promote long-time pressure tightness of threaded joints, the use of a material having lubricating, sealing and adequately stable properties for the intended service is recommended.

NOTE: Where PTFE sealing and anti-seize tape is used, care may be necessary to avoid rupture of thin-walled fittings.

Where parallel threads are used and sealing is to be made on a curved surface, a joint face shall be provided in accordance with Figure 3.19.4(a).

3.19.4.7 *Length of thread engagement*

The length of thread engagement—

(a) shall be in accordance with the appropriate Standard;

(b) shall develop adequate strength against blowout; and

(c) shall in no case have less than four pitches of complete thread (see also Clause 3.19.4.3).

3.19.4.8 *Attachment*

Screwed pipe and mountings may be attached to the vessel wall by screwing direct or by use of screwed sockets, nozzles, pad connections or socket welded connections.

Screwed sockets shall—

(a) have a body thickness measured at the major diameter of the thread in accordance with Clause 3.19.10.2(b);

NOTE: Additional thickness in the body of the socket should be provided to limit distortion during welding.

(b) meet the reinforcement requirements of Clause 3.18;

(c) be welded in accordance with Figure 3.19.4 (see Clause 3.19.3.5); and

(d) for connection involving Groups F and G steels, be made with full penetration butt welds and be connections of the following types only:

(i) For Group F steel—Figures 3.19.4(e), (f), (k) or (p)

(ii) For Group G steels—Figure 3.19.4(p).

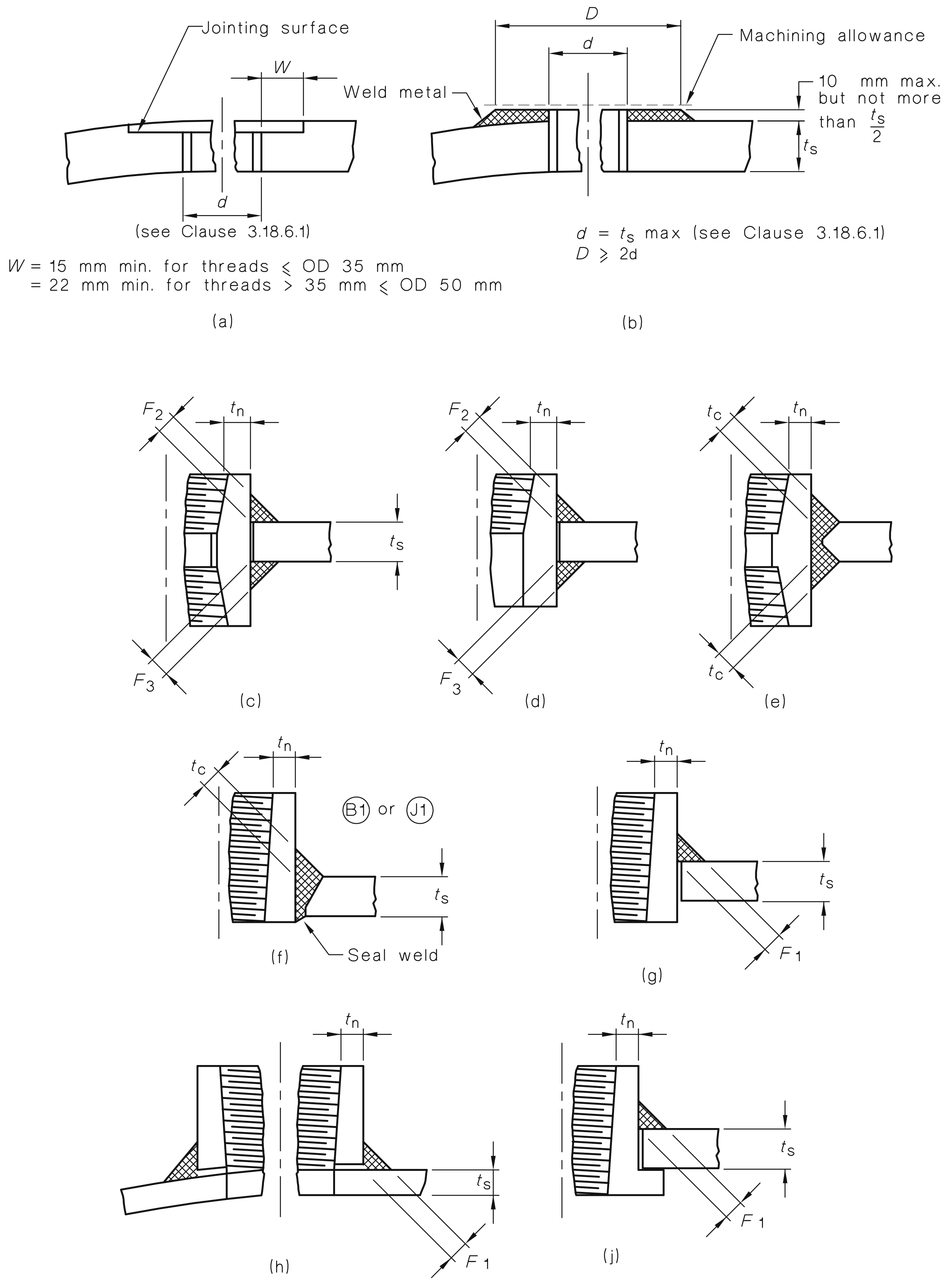
Where the plate thickness is insufficient to provide the specified length of thread engagement, a pad may be used. Pads of weld metal shall have a finished thickness not exceeding 50 percent of plate thickness with a maximum of 10 mm and the outside diameter should be approximately twice the hole diameter (see Figure 3.19.4(b)).

Screwed and socket welded connections shown in Figure 3.19.4(c), (d), (e), (f), (k), (l), (m), (n), or (o), not exceeding 90 mm outside diameter (i.e. DN80), may be attached by welds as follows:

(i) For the partial penetration welds or fillet welds, F_2 and F_3 shall not be less than the smaller of 2.5 mm, $0.7t_n$ and $0.7t_s$.

(ii) $(F_2 + F_3)$ shall not be less than the smaller of $1.25t_n$ and $1.25t_s$. Screwed nipples may be attached using welded connections similar to those in Figure 3.19.4, with the minimum thickness measured at the minor diameter of the thread.

Where welded pads are used, the attachment of the pads shall be in accordance with Clauses 3.19.3.



DIMENSIONS IN MILLIMETRES

FIGURE 3.19.4 (in part) SOME ACCEPTABLE SCREWED AND SOCKET WELDED CONNECTIONS

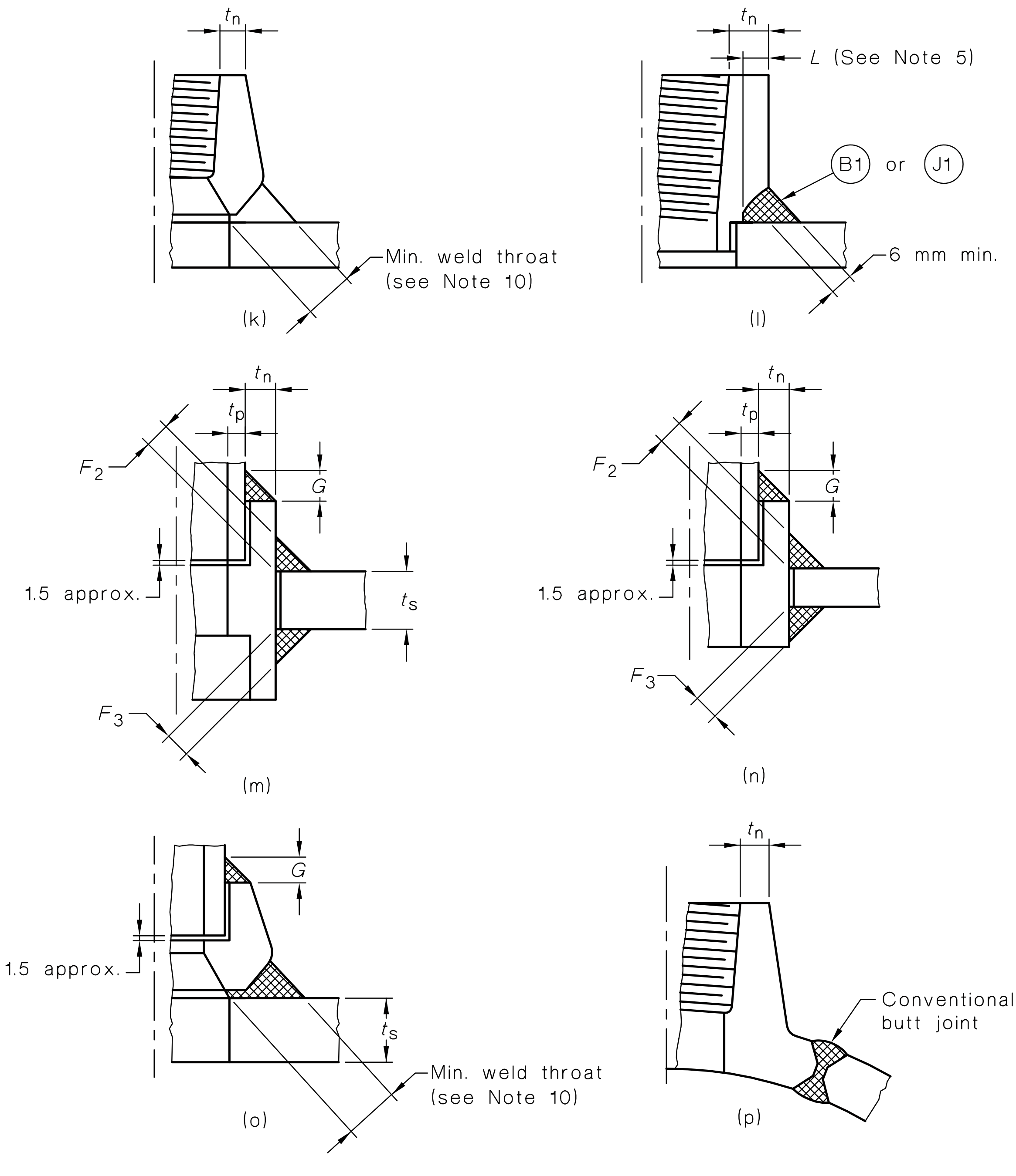


FIGURE 3.19.4 (in part) SOME ACCEPTABLE SCREWED AND SOCKET WELDED CONNECTIONS

NOTES TO FIGURE 3.19.4:

1 For weld preparations and weld sizes not shown, see Legend and Notes to Figures 3.19.3(A) to (D).

2 All screwed connections shall have the required length of thread.

4 In all set-on connections, the shell plate around the hole shall be examined visually for laminations prior to welding and where practicable, for lamellar tearing after welding.

5 L = thickness of ANSI B36.10 pipe of schedule 160 wall thickness and bore equal to nominal opening size.

6 See Figure 3.19.3(D) for standard nozzle weld details.

7 Size limitations are set out in Clauses 3.19.4.3 and 3.19.4.4.

8 In connections (m), (n), (o) diametrical clearances = 1 max. and $G = 1.25t_p$, but not less than 3 (see also Clause 3.19.4.4).

9 For all set-on connections and set-in partial penetration welds, see Clause 3.19.3.3 and AS 4458 for edge finish of unwelded opening.

10 The weld shall fill the weld preparation provided on the fitting unless otherwise specified by the designer.

11 Connection (l) is limited to fittings DN 80 or less in size.

12 For weld size exemptions, refer to Clause 3.19.4.8.

13 Connections (c), (d), (l), (m) and (n) shall not be used for Class 1H, 2H, 1S or 2S.

14 Connections (g), (h) and (j) are limited to Class 3 vessels with $t_s \leq 10$ mm.

3.19.5 Not allocated.

3.19.6 Studded connections

3.19.6.1 *General*

Studded connections may be used for jointing nozzles or fittings to the vessel wall either directly to a flat surface machined on the vessel wall or on a welded built-up pad or on a properly attached pad or fitting.

3.19.6.2 *Types of connection*

Some acceptable types of connections are shown in Figure 3.19.6, except that connections involving Groups F and G steels shall be attached with full penetration welds only in accordance with —

(a) for Group F steels—Figure 3.19.6(g), (h) or (j); and

(b) for Group G steels—Figure 3.19.6(g).

Where the connection is made direct to wall as shown in Figure 3.19.6(a) and (b), the diameter of the hole shall not exceed 75 mm.

It is recommended that dimensions comply with standard flange drillings and facings.

3.19.6.3 *Studs*

The material, dimension and number of studs shall comply with Clause 3.21.

3.19.6.4 *Stud holes*

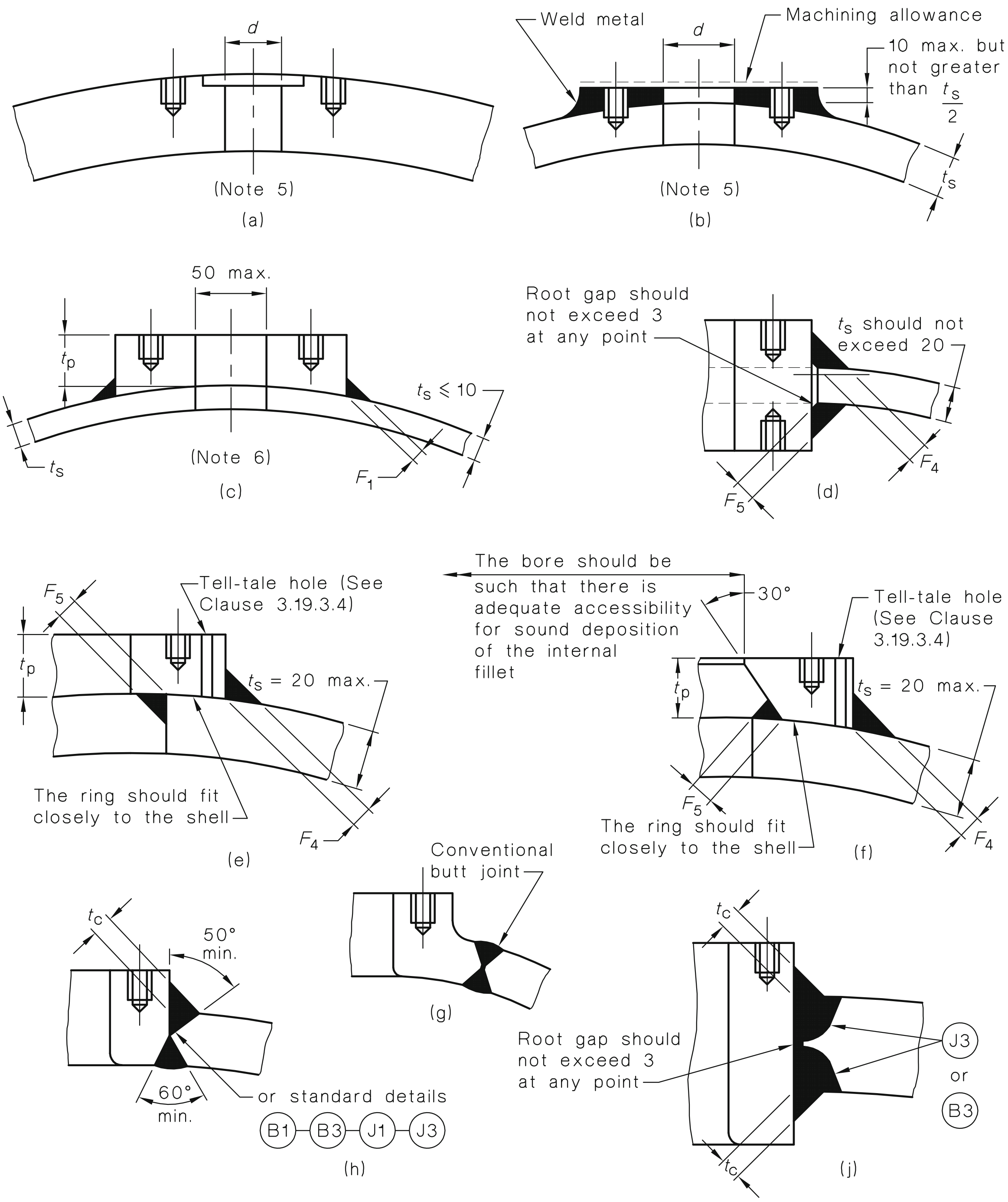
Stud holes in pad type openings such as those shown in Figure 3.19.6 (excluding (e) and (f)) shall be drilled without piercing the pressure-retaining surface unless the pierced surface is suitably sealed. The thickness below the unpierced hole shall exceed the required corrosion allowance by an amount sufficient to retain pressure and withstand piercing by the stud.

The sealant for a pierced surface shall not be corrodible by the contents of the vessel or shall be at least as thick as the thickness required under unpierced holes.

Stud holes in pad type openings such as those shown in Figures 3.19.6(c), (e) and (f) should not pierce the pad or ring. If they do, arrangement shall be made to prevent damage to attachment welds or the vessel wall due to tightening of the stud.

NOTE: Where bolts are fitted to pierced holes using a sealing compound that is compatible with the contents of the vessel and suitable for the service temperature range specified, no additional sealing is required.

The full threaded portion of stud holes and thread shall comply with Clause 3.21. For aluminium and its alloys, threaded inserts should be used where steel studs are used.



DIMENSIONS IN MILLIMETRES

FIGURE 3.19.6 SOME ACCEPTABLE STUDDED CONNECTIONS

NOTES TO FIGURE 3.19.6:

1 Connections (c), (d), (e), (f) are not recommended if the vessel is subjected to pulsating loads.

2 For weld symbols and weld sizes not shown, see Legend and Notes to Figures 3.19.3(A) to (C). Additionally, the sizes of fillet welds in (c), (d), (e) and (f) shall comply with the requirements of Clause 3.19.2.

3 To ensure gasket face flatness and alignment of tapered holes, consideration should be given to weld set-up, weld procedure and machining after fabrication, especially for connections (g), (h) and (j).

4 See Figure 3.19.3(D) for standard nozzle weld details.

5 Refer to Figures 3.19.4(a) and (b) for details.

6 Item (c) is permitted for Class 3 vessels only. This connection should not be used for corrosive conditions.

7 In all set-on connections, the shell plate around the hole shall be examined visually for laminations prior to welding and where practicable for lamellar type tearing after welding.

8 For all set-on connections and set-in partial penetration welds, see Clause 3.19.3.3 and AS 4458 for edge finish of unwelded opening.

9 t_s = nominal thickness of vessel wall minus corrosion allowance, in millimetres.

10 t_n may be taken as equal to t_p for the determination of weld size.

11 Connections (d), (e) and (f) shall not be used for Class 1H, 2H, 1S or 2S.

3.19.7 Expanded connections

3.19.7.1 *Application*

A pipe, tube or forging for connection of pipes and fittings to pressure vessels may be attached to the vessel wall by inserting through an opening and expanding into the wall, provided the outside diameter is not greater than 65 mm when fitted in an unreinforced opening, or not greater than 150 mm when fitted in a reinforced opening. For connections in flat tubeplates see Clause 3.17.

Expanded connections shall not be used as a method of attachment to vessels used for the processing or storage of flammable or toxic gases and liquids unless the connections are seal welded.

3.19.7.2 *Methods of attachment*

Such connections shall be—

(a) firmly expanded and beaded;

(b) expanded, beaded and seal-welded around the edge of the bead;

(c) expanded and flared not less than 2.5 mm over the diameter of the hole;

(d) expanded, flared and welded; or

(e) rolled and welded without flaring or beading, provided that—

   (i) the ends extend at least 6 mm but not more than 10 mm through the shell; and

   (ii) the throat of the weld is at least 5 mm but not more than 8 mm.

Where the outside diameter of the tube or pipe does not exceed 38 mm, the shell may be chamfered or recessed to a depth at least equal to the thickness of the tube or pipe and the tube or pipe may be rolled into place and welded. In no case shall the end of the tube or pipe extend more than 10 mm beyond the shell.

3.19.7.3 *Tube holes*

Tube holes shall meet the requirements of Clause 3.17.10.

Where the tubes are not normal to vessel wall, there shall be a neck or belt of parallel seating of at least 12 mm in depth measured in a plane through the axis of the tube at the holes.

3.19.7.4 *Expansion*

The expansion of tubes shall comply with Clause 3.17.11.

3.19.8 Brazed connections

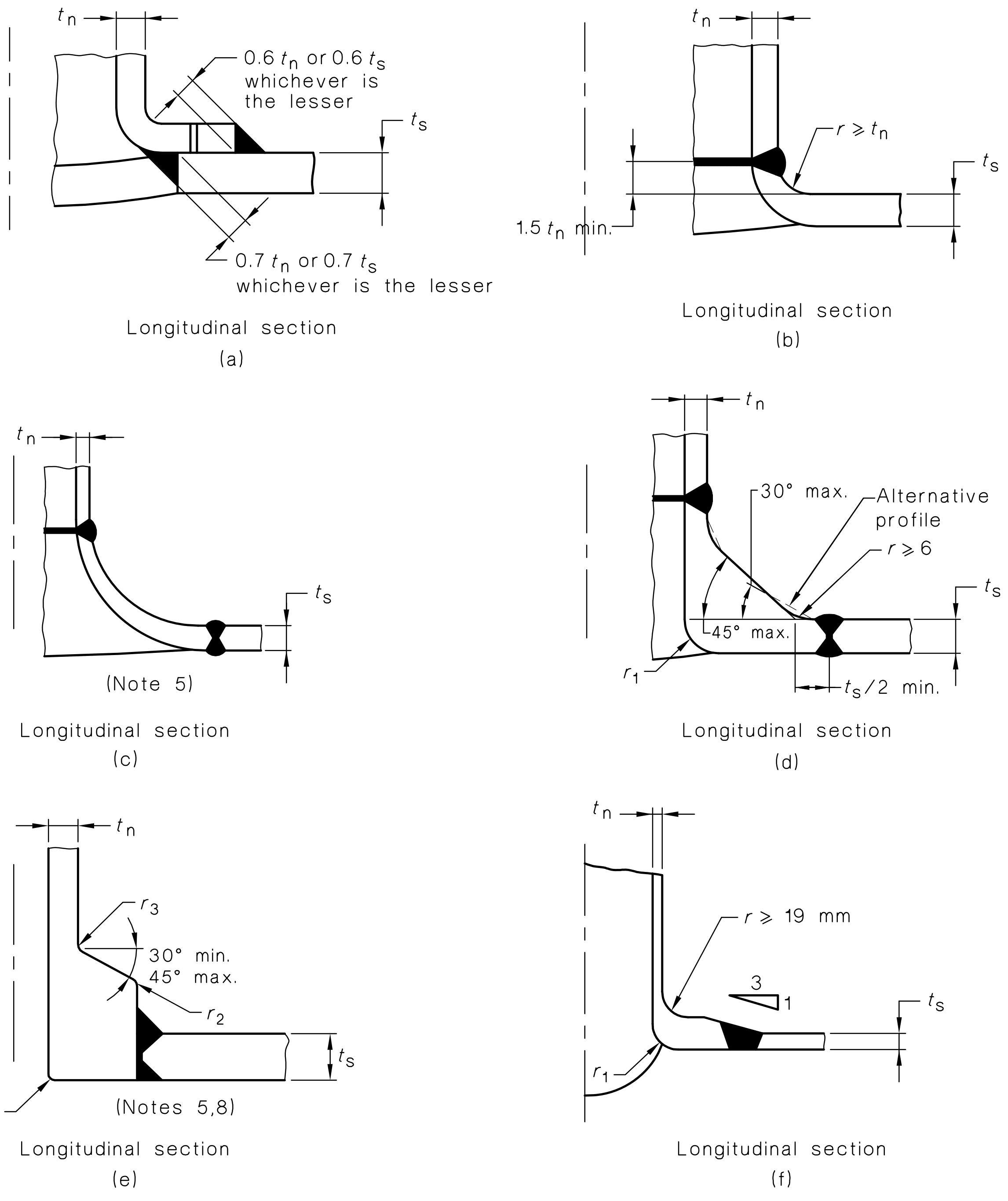
Connections such as saddle-type fittings and fittings inserted into openings formed by outward flanging of the vessel walls, in sizes not exceeding 90 mm outside diameter, may be attached to vessels by lap joints of brazed construction. Sufficient brazing shall be provided on either side of the line through the centre of the opening parallel to the longitudinal axis of the shell to develop the strength of the reinforcement as specified in Clause 3.19.2 through shear in the brazing.

Openings for nozzles and other connections shall be far enough away from any main brazed joint so the joint and the opening reinforcement do not interfere with one another.

3.19.9 Special connections

Methods of connections using forged components or forging a short nozzle in the vessel (see Figure 3.19.9) may be used.

For special connections through jacketed vessels see Clause 3.23.



DIMENSIONS IN MILLIMETRES

FIGURE 3.19.9 SOME ACCEPTABLE FORGED NOZZLE CONNECTIONS

NOTES TO FIGURE 3.19.9:

1 Conventional butt joints are used to connect the forging to the shell and nozzle and may therefore be of other forms than that shown.

2 These forgings connecting nozzles to shells are used with various forms of profile.

3 See Figure 3.19.3(D) for standard butt weld details.

4 t_n and t_s are nominal thickness minus corrosion allowance.

5 See Clause 3.19.3.2 for additional limitations when using Group F or Group G steels.

7 For extruded nozzles (i.e. items (a), (b) and (c)), radius is to be on inside and outside of vessel wall.

8 r_1 is at the inside corner of the nozzle for set-in connection, or the inside corner of the shell for set-on connection.

For Class 1, 2 and 3, r_1 is the lesser of 0.25t and 3 mm.

For Class 1H and 2H, $0.125t_s \le r_1 \le 0.5t_s$.

For Class 1, 2 and 3, $r_2 \ge 3$ mm.

For Class 1H, 2H, 1S and 2S, $r_2 \ge 6$ mm.

For Class 1, 2 and 3, $r_3 \ge 3$ mm.

For Class 1H, 2H, 1S and 2S, $r_3 \ge 20$ mm.

9 Connection (a) shall not be used for Class 1H, 2H, 1S or 2S.

3.19.10 Nozzles

3.19.10.1 *Design basis*

Nozzles shall be designed to provide—

(a) adequate thickness to withstand the design pressure and any corrosion;

(b) adequate thickness to withstand external loads when so specified by the purchaser (see Clause 3.2.3 and Appendix E);

(c) suitable connection to the vessel (see other requirements to this Clause (3.19)); and

(d) where necessary, added reinforcement of the opening.

3.19.10.2 *Nozzle thickness*

A2 The minimum thickness of the nozzle after fabrication, up to the connection to external piping shall be the greater of—

(a) the thickness to withstand both the calculation (internal or external) pressure and other loadings plus corrosion; and

(b) the smaller of—

(i) the required thickness of the vessel wall due to the larger of the design internal pressure and the design external pressure with this pressure applied as an internal pressure, at the point of attachment plus corrosion; and

(ii) the thickness given by $(D_o)^{1/4}$ plus corrosion, where D_o is the nozzle outside diameter, in millimetres.

The thickness required by Item (b) does not apply for access openings or openings for inspection only, or where suitable protection or support is provided. It is recommended that advantage be taken of increased nozzle thickness where reinforcement is required.

NOTE: Reinforcement of the hole in the shell of a vessel is obtained more efficiently by a thick nozzle pipe than by a thin one with a reinforcing ring.

3.19.10.3 *Inclination*

Nozzles may be inclined provided the reinforcement is adequate. This will be achieved by using the major dimension of the resultant opening in applying the requirements for reinforcement.

3.19.10.4 *Design for external loads*

Design of nozzles against external loads shall be according to Appendix N.

3.20 INSPECTION OPENINGS

3.20.1 General

All vessels, except those permitted by Clauses 3.20.5 and 3.20.6, shall be provided with suitable inspection openings to permit visual examination and cleaning of internal surfaces. Internal access equipment shall be provided where required. Consideration shall be given to the proposed method and frequency of, and accessibility for, inspection (see AS/NZS 3788).

Manholes shall be positioned to allow easy ingress by an inspector and safe and ready recovery of an incapacitated person.

3.20.2 Not allocated.

3.20.3 Not allocated.

3.20.4 General purpose vessels

Vessels other than special vessels covered by Clauses 3.20.5 and 3.20.6 shall be fitted with inspection openings in accordance with Table 3.20.4, or openings shall be located to permit closer inspection of areas most subject to deterioration.

TABLE 3.20.4

INSPECTION OPENINGS FOR GENERAL PURPOSE VESSELS

Inside diameter of vessel mm	Minimum clearance size of openings (Note 1) mm	Minimum number of openings (Note 2)	Location of openings
≤315	30 dia.	One for shells up to and including 900 mm long	In end, or where this is not practicable, in the shell near end.
		Two for shells over 900 mm long	—
>315 ≤460	40 dia.	Two for shells of any length	One in each end, or where this is not practicable, in the shell near each end
>460 ≤920 (Note 3)	50 dia.		
>920 ≤1500† (Note 3)	Handhole 150 diameter or 180 × 120	Two for shells up to 3000 mm long (Note 4)	One in each end or in the shell near each end
	Headhole 290 diameter	One for shells up to 3000 mm long (Note 4)	In the central third of the shell (Note 5)
>1500	Elliptical manhole or equivalent*	One for shells of any length	In the shell or end to give ready ingress and egress

* See Table 3.20.9.

† Either handhole or headhole option may be selected.

NOTES:

1 Size openings for jackets of jacketed vessels need not exceed 65 mm OD.

2 The length of shell is measured between the welds attaching the ends to the cylindrical shell.

3 Inspection, head-and hand holes may be omitted if a manhole is provided.

4 For shells longer than 3000 mm, the number of openings shall be increased so the maximum distance between handholes does not exceed 2000 mm and that of headholes 3000 mm.

5 For shells up to 2000 mm long, a single headhole in one end may be used.

3.20.5 Vessels not subject to corrosion

Vessels that are not subject to internal corrosion, abrasion or erosion; and which are—

(a) used for static services (i.e. stationary, or normally stationary but are transported at infrequent intervals, and are not subject to severe shock or fatigue loadings), and have a water capacity not exceeding 60 m^3;

(b) used for other than static service, and have a water capacity not exceeding 5 m^3; or

(c) buried or mounded, and have a water capacity not exceeding 15 m^3;

shall be fitted with inspection openings in accordance with Table 3.20.5. Vessels exceeding the limits of Item (a) or Item (b) shall be fitted with a manhole, except where the process or nature of the vessel contents or the design of the vessel renders undesirable the installation of a manhole. For vacuum-insulated vessels where a manhole is provided in the inner vessel, but not in the outer vessel, the manufacturer shall clearly mark the outer vessel with the words ‘Manhole Here’ opposite the inner manhole.

For the purposes of this Standard, vessels considered not subject to corrosion include those containing refrigerants, liquefied petroleum gas and other substances that are known to have no harmful effect on the vessel material.

TABLE 3.20.5

INSPECTION OPENINGS IN VESSELS NOT SUBJECT TO CORROSION

Inside diameter of vessel **mm**	**Minimum clearance size of opening, mm** **(Notes 2 and 3)**	**Minimum number and location of openings** **(Note 1)**
≤160	Not required	—
>160 ≤250	25	For shells ≤3000 mm: one opening in end (or in shell near end) For shells >3000 mm: two openings: one in each end (or in shell near each end)
>250 ≤400	30	
>400 ≤775	35	
>775	40	

NOTES:

1 A greater number of smaller openings may be used provided—

(a) the minimum opening is 25 mm clearance diameter;

(b) the sum of the diameters is at least equal to that required by Table 3.20.5;

(c) the openings are suitably located to facilitate inspection.

2 These openings may be provided by—

(a) removal of valves, fittings or pipe;

(b) cutting of nozzle pipes near the shell;

(c) special inspection nozzles with seal welded closures.

3 As an alternative to openings in these vessels, inspection may be provided by—

(a) cutting of shells;

(b) having no openings and using non-destructive examination methods (other than visual), see Clause 3.20.6(b). Refer to Appendix E.

3.20.6 Vessels not requiring openings

Vessels need not have the prescribed inspection openings where—

(a) they are designed, constructed and installed so as to be readily dismantled to permit visual examination and cleaning of all internal surfaces subject to stress; or

(b) they are of such design and usage that visual examination is not practicable and an alternative approved means of assessing deterioration is applicable.

3.20.7 Manholes for vessels containing an unsafe atmosphere

Vessels that, at the time a person may be required to enter the vessel, are liable to contain an unsafe atmosphere, i.e. contaminated or oxygen deficient, shall be fitted with at least one manhole of the following minimum size:

(a) For stationary vessels—not less than 450 mm × 400 mm (elliptical) or 450 mm (circular).

(b) For transportable vessels—not less than 400 mm × 300 mm (elliptical) or 400 mm (circular).

NOTE: The means of ingress to and egress from a vessel need to be kept free from encumbrances. Accordingly, where the atmospheric contaminants or the nature of the work to be performed in a vessel may require power lines, hoses, ventilation ducts or similar to pass through a manhole, consideration should be given to the provision of a second manhole. (See also AS 2865 and AS/NZS 3788.)

3.20.8 Alternative openings

Openings may alternatively be provided as follows:

(a) Where the vessel shape is not cylindrical, the openings required by Clause 3.20.4 need not apply, but sufficient openings of suitable size and location shall be provided to give access equivalent to that required by this Clause (3.20).

(b) Where a manhole is prescribed but the shape or use of the vessel makes the provision of manhole impracticable, sufficient inspection openings each 150 mm × 100 mm or 125 mm diameter, or larger, shall be provided. One opening shall be located in each end or in the shell near the end and in other positions to permit examination of all areas liable to deterioration.

(c) Vessels that are 350 mm inside diameter or less may use pipe or fitting connections with a clearance size of opening greater than or equal to 19 mm in place of the required inspection openings, provided they are in suitable locations, can be readily dismantled and will provide the necessary number and size of openings.

(d) Core openings in cast vessels having internal passages may be used in place of inspection openings, provided their closures are readily removable and replaceable and are located to permit adequate inspection.

(e) Removable ends or cover plates may be used in place of inspection openings, provided they are at least equal to the minimum size required for such openings. A single removable end or cover plate may be used in lieu of all other inspection openings where the size and location of such an opening permits a general view of the interior at least equal to that obtained with the inspection openings otherwise required.

3.20.9 Size of openings

The preferred sizes of inspection openings are given in Table 3.20.9.

A manhole should not be smaller than 500 mm clear opening diameter unless agreed between the parties concerned.

TABLE 3.20.9
SIZE OF INSPECTION OPENINGS

millimetres

Type	Circular openings (see Note 1)	Equivalent elliptical openings— (major × minor axes)	Maximum depth of opening (see Note 2)
Sighthole	50	—	50
	40	—	40
	30	—	30
Handhole	200	225 × 180	100 Pads
	150	180 × 120	75 Pads
	125	150 × 100	63 Pads
	100	115 × 90	50 Pads
	75	90 × 63	50 Pads
Headhole	300 max.	320 × 220 max.	100
	290 min.	310 × 210 min.	
Manhole (see Note 3)	550 (Preferred)	—	500
	500 (Min. recommended)	—	300
	450 (Exceptional)	450 × 400	245

NOTES:

1 Circular opening is the clear inside diameter, and is not equivalent to the DN.

2 The depth of the opening is the shortest distance from the outside face of the opening to the inside face of the opening. Linear interpolation of the depth of the opening is permitted. A greater depth may be permitted only where the tabulated depth is impractical.

3 A 400 mm × 300 mm elliptical manhole or 400 mm diameter circular manhole may be used only where larger manholes are impractical and within the following limitations (see also Clause 3.20.7(b)):

(a) Vessels for steam, water, air or other applications where it can be ensured that, at the time of any entry to the vessel, the contents will be life supporting.

(b) For stationary vessels, the diameter of the vessel is not greater than 1530 mm.

(c) For horizontal vessels, the manhole is in the end with the major axis of the ellipse horizontal. (See also Clause 3.20.1).

(d) For vertical vessels, the manhole is in the shell within 700 mm to 900 mm of the bottom of the vessel or a platform within the vessel, and with the major axis of the ellipse horizontal.

(e) Maximum depth of opening is 150 mm.

4 For guidance on working in confined spaces, refer to AS 2865.

3.20.10 Design of inspection openings

The design of inspection openings shall be in accordance with the requirements for openings and nozzles (see Clauses 3.18 and 3.19).

NOTE: For the design of viewports, see Clause 3.2.9.

Screwed plugs with parallel threads may be used for closure of inspection openings up to and including 65 mm outside diameter, provided they have a joint face and the connection is in accordance with Clause 3.19.3 or approved method that will permit removal and replacement from time to time with ease and safety. The screw plug shall be of material suitable for the pressure and temperature conditions.

3.20.11 Ingress to vessels

Unless precluded by process equipment or other circumstances, provision shall be made inside the vessel so that a safe landing place or the top rung of a ladder leading thereto is located adjacent to and not more than 1 m below any manhole intended for ingress. Suitable hand grips shall be provided where practicable.

3.21 BOLTED FLANGED CONNECTIONS

3.21.1 General

This Clause (3.21) deals with the design of bolted flanged connections in pressure vessels, including inspection covers, blind flanges, bolted flat ends, sections of shells, and nozzle connections. The design requirements provide for hydrostatic end-forces, moments or forces due to external loading, and gasket seating.

Bolted flanged connections shall comply with one of the following:

(a) Design by calculation using the requirements of this Clause (3.21), using stresses within the limits for Class 1 vessels.

NOTE: More conservative design strengths should be considered for flanges exceeding 1000DN, or where flange face finish, flatness, or bolt tightening practices may not be of high quality, or where risk of leakage cannot be tolerated.

(b) Design by finite element analysis (FEA), according to the requirements of Appendix I, and using stresses within the limits of Appendix A, B or H.

(c) One of the following Standards, within the scope of application of that Standard:

(i) ASME B16.5.

(ii) ASME B16.47 Series A.

(iii) AS 2129, within the limits of Clause 3.21.2(b), and not for lethal contents or where risk of leakage cannot be tolerated.

NOTE: There is the possibility of high stresses and leakage with these full-faced flanges when used at the maximum conditions permitted by that Standard, particularly under hydrostatic test.

(iv) AS/NZS 4331 or ISO 7005.

(v) EN 1092.

The design calculations in this Clause (3.21) or other design evaluation to prove the adequacy of the flange design need not be carried out for bolted flange connections that comply with Items (c)(i), (ii), (iv) or (v) above, subject to the following:

(A) Flanges shall be used within the size and pressure-temperature ratings permitted by the nominated Standard.

(B) Gasket materials shall be within proven industry practice.

(C) The flange connections are not subject to significant external loading.

A2 For the purpose of this Clause, significant external loading is considered to be a combination of design pressure, external loads and external moments that, when converted to an equivalent pressure as per Equation 3.21.6.4.1(1), are greater than 150% of the flange rated design pressure at design or operating temperature. For the determination of P_e as per Equation 3.21.6.4.1(1), the pressure term 'P' shall be less than or equal to the flange rated pressure per its nominated design code. Where external loading plus design pressure exceeds the 150% value, the designer shall consider the need for further evaluation based on known operating experience, consequences of a leak, conservatism of design loading, calculated percentage of flange rated pressure and other relevant influences.

Bolted flange connections that comply with Items (a), (c)(iii) or (vi), and are larger than DN600 or are subject to a specified or significant external load, shall also comply with Clause 3.21.6.8.

Bolted flange connections that comply with Item (b) shall comply with Clause 3.21.6.8.

NOTE: Large diameter flanges have potential for leakage due to excessive rotation, particularly where sealed with non-self-energizing gaskets and placed under cyclic loading. European flange Standards also have potential for leakage, as they generally use lower safety factors than ASME and Australian Standards.

Non-circular flanges shall be designed in accordance with Clause 3.1.3(b) or (c). Calculations for gasket seating and bolting shall follow the principles of Clause 3.2.1. It is recommended that design account for flange deflection and rotation.

NOTE: Guidance on design of such flanges can be found in Enquiry Case 5500/133: September 2003, to PD 5500:2003.

3.21.2 Types of flanged connection

For design purposes flanges are divided into the following types:

(a) *Narrow-face flanges*—flanges in which the joint ring or gasket does not extend beyond the inside of the bolt holes as shown in Figure 3.21.2(b), (c), (d).

(b) *Flanges with full-face joint*—flanges in which the joint ring or gasket extends over the full width of the flange face as shown in Figure 3.21.2(a). These are suitable only when used with comparatively soft jointings and are not recommended for pressures exceeding 2.1 MPa or temperatures exceeding 260°C. Where the inside diameter exceeds 600 mm, a maximum of 1.4 MPa is recommended.

(c) *Reverse flanges*—flanges where the shell is attached to the outer edge of the flange as shown in Figure 3.21.12.2.

NOTE: Care must be taken to avoid the incorrect mating of full-face flanges with narrow-face flanges, particularly where the full-face flange is made from weaker material, e.g. standard full-face cast iron flanges should not be connected to standard narrow-face steel flanges.

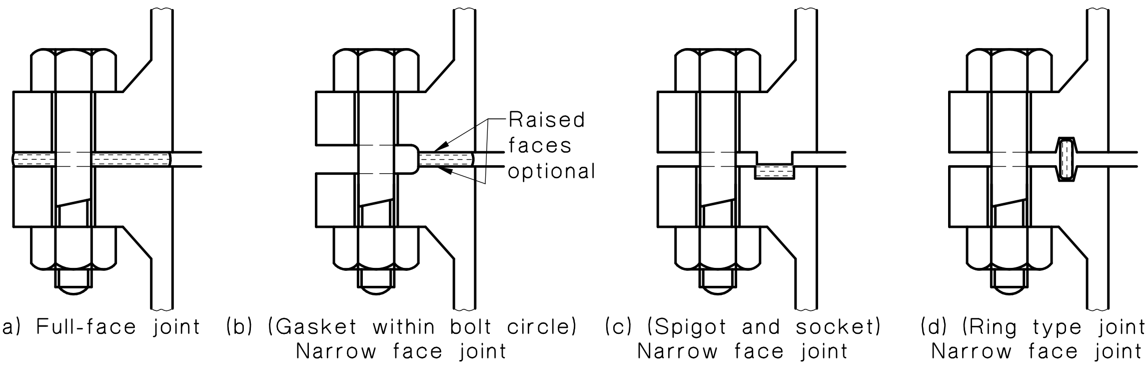


FIGURE 3.21.2 TYPES OF BOLTED FLANGE JOINT

3.21.3 Attachment of flanges

3.21.3.1 *Types of attachment*

Various types of flange attachment are shown in Figure 3.21.3.

Typical flange attachments, as shown in Figure 3.21.3, may be used for reverse flanges, adjusted as appropriate for the location of the flange inside the shell.

NOTE: Such flanges normally have provision for studs in place of bolt holes.

3.21.3.2 *Strength of attachment*

Flanges shall be attached to the vessel or nozzle in accordance with the dimensions shown in Figure 3.21.3.

Threaded flanges where used shall be provided with ample depth and length of thread to withstand loads and moments, and shall be screwed home hard on the nozzle pipe or vessel shell. Threads on the nozzle pipe or vessel shall vanish at a point just inside the back or hub of the flange, except where a parallel-to-parallel threaded attachment is used. In this latter attachment ample provision shall be made for sealing and locking.

3.21.3.3 *Limits of use of welded flange attachments*

Welded flange attachments shown in Figure 3.21.3 are limited to the following maximum calculation pressures and temperatures:

(a) Attachments (a) to (d) inclusive and (1)—no limits provided that full penetration welds are used for Group F and Group G steels.

(b) Attachments (e) to (g) inclusive—8.3 MPa at 50°C for carbon steel and equivalent ratings except temperature shall not exceed 425°C, e.g. Table R of AS 2129.

(c) Attachments (h) and (j)—4.9 MPa at 50°C for carbon steel and equivalent ratings; except that the temperature shall not exceed 425°C, e.g. Table J of AS 2129.

(d) Attachments (b), and (d) to (j) inclusive shall be avoided where thermal gradient may cause overstress in welds or where many large temperature cycles are expected, particularly where the flange is uninsulated.

(e) Attachments (e), (h) and (j) are not recommended for corrosive conditions.

(f) Slip-on and socket-welded flanges are not recommended for service below −45°C.

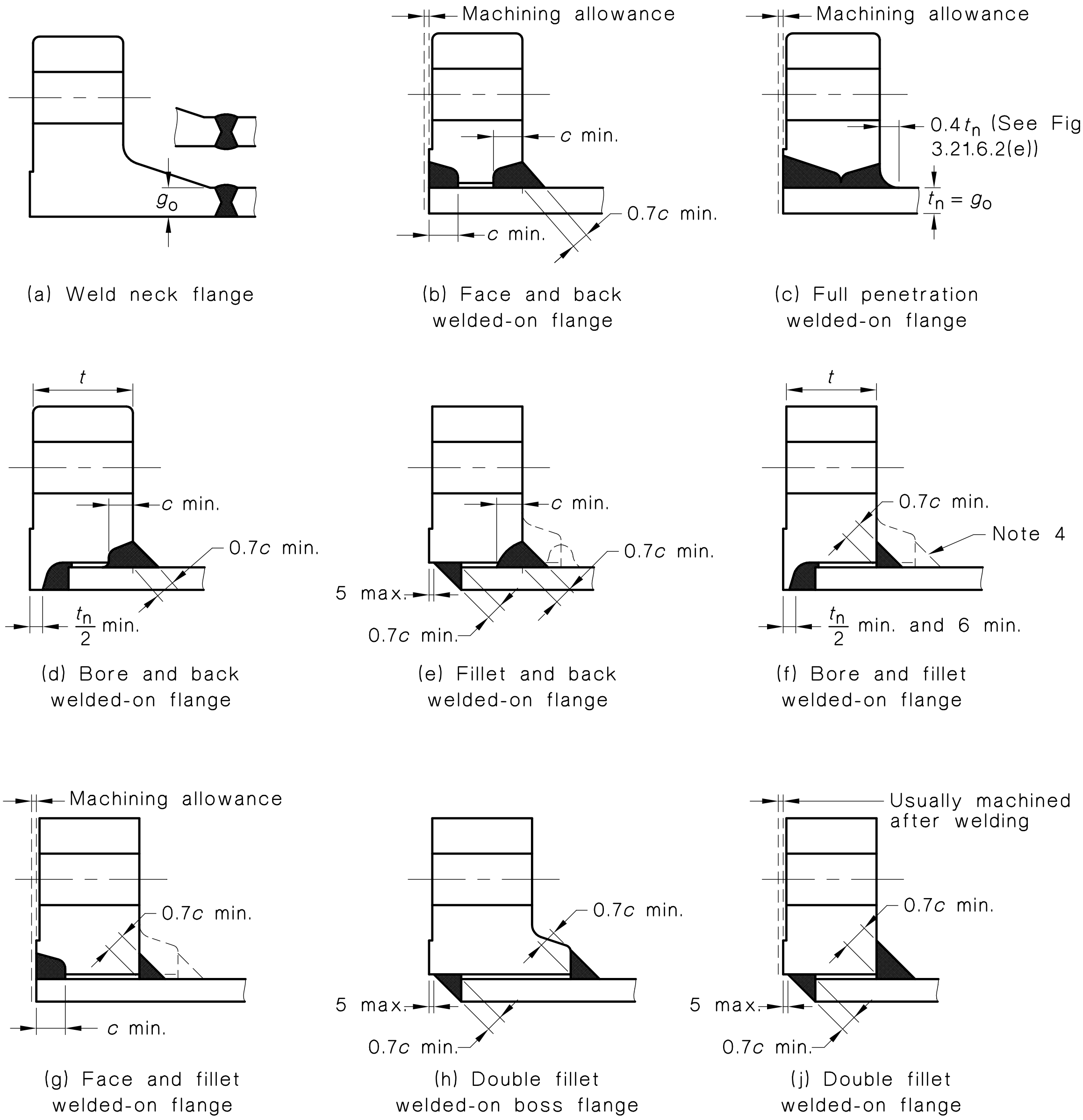
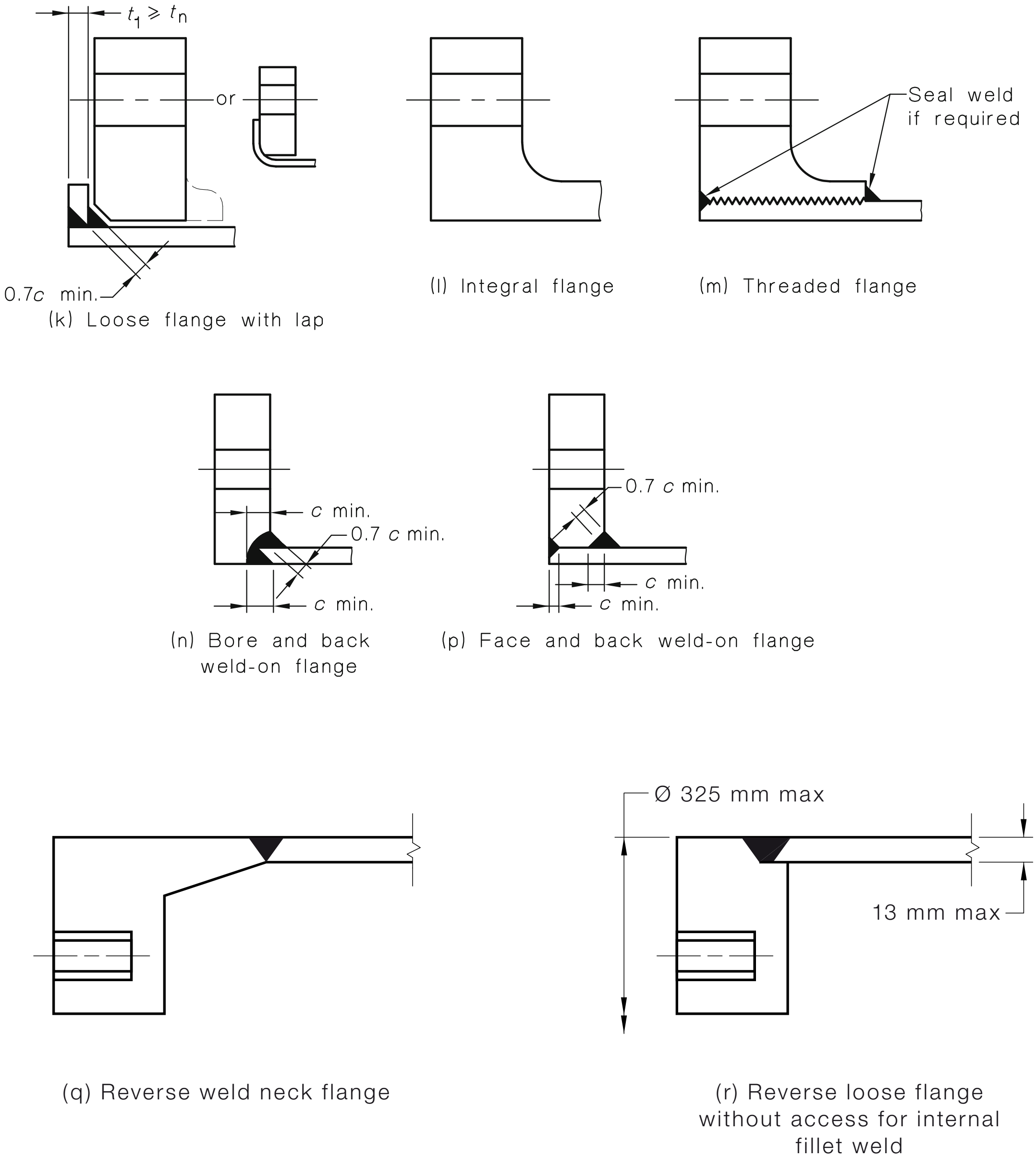


FIGURE 3.21.3 (in part) TYPICAL FLANGE ATTACHMENTS



NOTES:

1 $c = t_n$ or t_x, whichever is less, where t_x is as in Clause 3.21.6.2.

2 t_n = nominal thickness of shell or nozzle, less corrosion allowance.

3 See Figure 3.19.3 for standard weld preparations. Where weld preparations are shown as J type on this Figure (3.21.3), B type preparations may be used.

4 The gap between flange and wall should not exceed 3mm. Wide gaps increase the tendency to cracking during welding, particularly as the thickness of the parts joined increases. For welding of thin sections by gas tungsten-arc welding (GTAW) process, the gap is to be small.

5 For thin shells it may be desirable to fit a short length of thickened shell to facilitate flange attachment.

6 For flange dimensions not shown in Items (a) to (p), see Figure 3.21.6.2, for Items (q) and (r) see Figure 3.21.12.2.

FIGURE 3.21.3 (in part) TYPICAL FLANGE ATTACHMENTS

3.21.3.4 *Limits of use of threaded flanges*

The pressure and temperature limits of use of threaded flanges in which the tightness of the joint depends upon the tightness of the threads are given in Table 3.21.3.4.

Threaded flanges are not recommended for severe cyclic service, or for corrosive service unless sealed on the face, or for service below −50°C for ferritic steels.

Threaded flanges in which the tightness of the joint depends on the tightness of the threads and which contain—

(a) corrosive material; or

(b) flammable or toxic fluids or fluids that are difficult to contain;

shall be of weldable quality and shall be seal welded. In the case of (a) the seal weld shall be on the contact face, and in the case of (b) on the back of the flange.

3.21.4 Basis of design

3.21.4.1 *General*

The design of a bolted flanged connection involves the selection of gasket (material, type and dimensions), flange facing, bolting, hub proportions, flange width and flange thickness. The methods given in the following clauses generally require a preliminary selection of these details followed by a trial and error procedure. Once the connection is designed on the basis of stress limits, it is then checked for serviceability by determining joint rigidity and flanges stress under actual bolt-up conditions.

Notation for this Clause (3.21.4) is shown in Clause 3.21.6.2.

Flange dimensions shall be such that the stresses in the flange calculated in accordance with this Clause (3.21.4) do not exceed the allowable flange stresses permitted in Clause 3.21.6.7. All calculations shall be made with dimensions in the corroded condition, i.e. by allowing for loss of metal from all 'wetted' surfaces equal to the corrosion allowance.

In the design of a bolted flange connection, complete calculations shall be made for two separate and independent sets of conditions, which are defined in Clauses 3.21.4.2 and 3.21.4.3.

NOTE: It is recommended the bolt spacing on unproven flange designs be not greater than the dimension obtained from the following equations (see Clause 3.21.6.2 for notation):

$$P_b \text{ max.} = 2D_b + \frac{6t}{m+0.5} \text{ for narrow face flanges; or}$$

$$P_b \text{ max.} = 2D_b + \frac{6t}{m+0.5}\left(\frac{E}{200\,000}\right)^{0.25} \text{ for flanges with full face gaskets.}$$

If bolt spacing, P_b exceeds $2D_b + \frac{6t}{m+0.5}$, the total flange design moments (including external loads, additional gasket areas, and other relevant influences) should be increased by a factor equal to

$$\left(\frac{P_b}{\left(2D_b + \frac{6t}{(m+0.5)}\right)}\right)^{0.5}$$

The minimum bolt spacing should consider the space necessary to apply a wrench to the nuts and possible interference from gussets and other obstructions.

TABLE 3.21.3.4

LIMITS OF USE OF THREADED FLANGES

Material	Method of attachment	Maximum pressure MPa	Maximum temperature °C
Carbon and carbon-manganese steels	Threaded and expanded Taper to taper Taper to parallel	3.1 2.1 0.86	371 260 260
Alloy steels	Threaded and expanded	4.2	482
Cast irons	Taper to taper Taper to parallel	1.05 0.86	180 178
Copper and copper alloys	Threaded	Refer to AS 2129	

3.21.4.2 *Operating conditions*

The operating conditions shall be taken as those required to resist the combined load of the hydrostatic end-force of the design pressure, and any external loads, tending to part the joint, and to maintain on the gasket or joint contact surface sufficient compression to ensure a tight joint at the design temperature. The minimum force is a function of the design pressure, any external loads, the gasket material, and the effective gasket or contact area to be kept tight under pressure, as determined by Equation 3.21.6.4.1(2) or (3), and determines one of the two requirements for the amount of bolting area A_{m1}. This force is also used for the design of the flange, as determined by Equation 3.21.6.4.4(1). Where a gasket has surface additional to the calculated peripheral area (e.g. heat exchanger pass partitions), the total gasket area shall be included in Equation 3.21.6.4.1(2) or (3).

3.21.4.3 *Gasket seating conditions*

The gasket seating conditions shall be taken as those existing when the gasket or joint-contact surface is seated by applying an initial force with the bolts when assembling the joint, at atmospheric temperature and pressure. External weight-type loads applied to the flange assembly prior to pressurization may be excluded at the designer's discretion. The minimum initial force considered to be adequate for proper seating is a function of the gasket material, and the effective gasket or contact area to be seated (including any pass partition gasket area), as determined by Equation 3.21.6.4.1(4), and determines the other of the two requirements for the amount of bolting area A_{m2}. For the design of the flange this force is modified in accordance with Equation 3.21.6.4.4(2), to take account of the actual bolt up conditions, when these govern the amount of bolting area required, A_m, as well as the amount of bolting area actually provided, A_b.

3.21.5 Materials and components

3.21.5.1 *General*

Materials for bolted flanged connections shall comply with the requirements of Table B1 and Clause 2.2 (as applicable).

3.21.5.2 *Flange materials*

Where the thickness of the flange section before machining exceeds 75 mm, flanges made from ferritic steel and designed in accordance with this Clause (3.21.5) shall be given a normalizing or full-annealing heat treatment.

Material on which welding is to be performed shall be proven to be of good weldable quality. Satisfactory qualification of the welding procedure in AS/NZS 3992 is considered as proof. Welding shall not be performed on steel that has a carbon content greater than 0.35%. All welding on flange connections shall comply with the requirements for postweld heat treatment given in AS 4458.

Fabricated hubbed flanges may be machined from a hot-rolled or forged billet or forged bar. The axis of the finished flange shall be parallel to the long axis of the original billet or bar. (This is not intended to imply that the axis of the finished flange and the original billet must be concentric).

Hubbed flanges (except as permitted above) shall not be machined from plate or bar stock material unless the material has been formed into a ring, provided that:

(a) In a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange

NOTE: This is not intended to imply that the original plate surface be present in the finished flange.

(b) The joints in a ring are full-penetration welded butt-joints that conform to the requirements of this Standard. The thickness to be used to determine stress-relief and radiography shall be the lesser of:

$$t \text{ and } \frac{(A-B)}{2} \quad \text{. . . 3.21.5.2}$$

where these symbols are as defined in Clause 3.21.6.2.

(c) The hot or cold forming complies with the relevant Clauses in AS 4458.

(d) To ensure they are free from defects, the back of the flange and the outer surface of the hub are inspected (after any forming and subsequent heat treatment is completed) by non-destructive methods such as magnetic particle inspection or penetrant flaw inspection.

3.21.5.3 *Flange face surface*

Gasket contact face flatness has a significant influence on the leak-tightness of a joint.

For a machined flange, the maximum deviation from a plane on peripheral gasket contact face surfaces (i.e. the face flatness) should be one of the following:

(a) No greater than the lesser of 0.5 mm, and (gasket thickness/6).

(b) For Hazard Level A vessels (as per AS 4343), or where the purchaser indicates that high safety risks could result from leakage, within the tolerance shown below:

Flange ID mm	Tolerance from a plane			
≤400	Larger of:	±0.08 mm	or	(gasket thickness/15)
>400 ≤800	Larger of:	±0.15 mm	or	(gasket thickness/15)
>800 ≤2000	Larger of:	±0.20 mm	or	(gasket thickness/15)
>2000	Larger of:	±0.30 mm	or	(gasket thickness/10)

NOTES:

1 Gasket thickness refers to the compressible thickness, for example, a gasket with a metal core covered both sides with 0.5 mm thick graphite would have a gasket thickness of 1 mm for the purpose of these guidelines.

2 An accurate check may be made by relating the gasket manufacturer's recovery (as a percentage of the thickness compressed) to the allowable flange rotation. Allowable rotation should be limited to 0.3° for integral flanges and 0.2° for loose flanges. The use of (gasket thickness/6) indicates a recovery of 16.7%.

The maximum deviation should not occur in an arc of 20° or less. Face flatness measurements should be confirmed after welding or post-weld heat treatment. The use of a straight edge for measuring is acceptable.

For multi-pass heat exchangers, consideration should be given to pass partition gasket contact face flatness after fabrication. Where pass partition gaskets are substantially non compressive, they shall not interfere with the periphery gasket compression.

Designers should note that gasket contact face surface finish and bolt tightening procedures may also have a profound impact on joint sealing. Contact surface finish shall follow the recommendations of the gasket manufacturer. Bolt tightening procedures should be developed to achieve a gradual, uniform bolt load.

3.21.5.4 *Bolting*

3.21.5.4.1 *General*

Material used for bolting shall be suitable for use at all temperatures and conditions of the intended service.

Precautions shall be taken to avoid over-stressing of small diameter bolting during tightening and to avoid binding of threads. To prevent over-stressing, torque spanners or other means should be used for bolting up to 38 mm. To prevent binding, nuts and bolts should be lubricated on assembly, and manufactured from different materials or from materials having differing levels of hardness. See Appendix D for guidance on corrosion of dissimilar metals.

NOTE: ASME BPV-VIII-1 Appendix S gives guidance on approximate bolt loads achieved using manual tightening. See also Clause 3.21.6.4.1.

Bolting materials of a similar nature to the flange material and of adequate strength are likely to avoid corrosion difficulties.

3.21.5.4.2 *Bolts, screws, studs, stud-bolts and clamp bolts*

The following requirements shall be met:

A1 (a) Materials for bolts (including screws, studs, stud-bolts and clamp bolts) shall comply with the specifications listed in Table B2, Appendix B (which also gives design strengths). Additional materials or grades specifically listed as bolting material in ASME BPV-VIII may be used, provided design stresses determined are in accordance with this Standard.

 Alternatively, other non-bolting materials to recognized international Standards may be used, provided—

 (i) the selected material Standard lists yield and tensile strength at the design temperature of the equipment:

 (ii) the design strength is calculated in accordance with Table A1, Appendix A;

 (iii) the grain direction of the material is parallel to the axis of the fastener; and

 (iv) nuts meet the requirements of Clause 3.21.5.4.3 and load tested in accordance with the requirements of a nut material Standard listed in Table B2, Appendix B.

(b) When bolts are machined from heat-treated, hot-rolled, or cold-worked material and are not subsequently hot-worked or annealed, design strength shall be based on the condition of the material selected (see Table B2).

(c) When bolts are fabricated by hot-heading, the design strength for annealed material in Table B2 shall apply unless the manufacturer can furnish adequate control data to show the tensile properties of hot-rolled bars or hot-finished forgings are being met, in which case the design strength for the material in the hot-finished condition may be used.

(d) When bolts are fabricated by cold-heading, the design strength for annealed material in Table B2 shall apply unless the manufacturer can furnish adequate control data to show that higher design strengths, as agreed upon, may be used. In no case shall such stresses exceed the design strength given in Table B2 for cold-worked bar stock.

(e) Bolts and studs shall be suitably protected from corrosion.

NOTE: Where some corrosion is possible an increase in size or change in material is recommended.

(f) All bolts shall be made forged, pressed or machined from one piece, except that for ‘T’ or eye bolts the bolt may be welded to the cross part or eye, provided the material has good weldability, the bolt is normalized after welding, and the joint is a complete penetration weld, fully radiographed.

(g) Where metal temperatures exceed 400°C stud-bolts shall be used. Such stud-bolts shall be threaded full length or with the unthreaded portion reduced to the bottom of the thread. The surface finish shall be at least equal to R_a 0.8 μm.

(h) High strength and alloy bolts shall be marked to facilitate material identification.

(i) Clamp bolts shall be permanently attached to the vessel. Sample clamp bolt assemblies shall be tested to confirm R_e and R_m values satisfy the design strengths required for both. Clamp assemblies shall have the appropriate load carrying capacity and shall be distributed around the flange as specified in the design.

For swing bolts, see also Clause 3.27.3.

3.21.5.4.3 *Nuts*

Nuts shall be suitable for the service conditions and made of a material compatible with the bolts used.

Carbon-molybdenum steel nuts Grades 4 and 4B to BS 4882 and equivalent ASTM materials shall not be used at operating temperatures exceeding 600°C nor below −29°C. Carbon-molybdenum steel nuts that meet the impact test requirements of the Standards listed in Table B2 may be used at temperatures down to −100°C.

Nuts may be made of material to the same specification as for the bolting to which they are fitted, but for operating temperatures over 290°C the hardness (or strength) of ferritic nuts should differ from that of the bolts.

Nuts shall be of standard design but may be of any practical shape including those with holes or ring for ‘tommy bar’ or similar, provided the depth of the threaded portion is not less than the diameter of the thread and shall engage the bolt or stud by at least the same amount.

The thrust face of nuts shall be machined where the operating temperature exceeds 400°C.

Cap or blind nuts shall have a threaded portion with a length not less than 1.5 times the diameter of the thread.

Nuts for swing bolt closures shall have thread engagement as per Clause 3.21.5.4.7.

Nuts requiring welded attachments shall be made of suitable material with good weldability. Thread fit and condition of the nuts after welding shall comply with the relevant Standard.

3.21.5.4.4 *Washers*

The use of washers is optional. When used they shall be of wrought materials and shall be of approximately the same hardness and composition as the nuts when used with carbon or alloy steel bolts and studs.

3.21.5.4.5 *Threads*

Threads on bolts that are to be frequently unscrewed should be trapezoidal. For such bolts the length of thread under the nut shall be not less than 1.5 times the diameter of the thread.

For bolts with full shank the length of thread under the nut should be at least equal to the bolt diameter. Threads should be rolled, rather than cut, wherever possible. The bolt and nut combination shall be a medium or closer fit.

3.21.5.4.6 *Size*

Bolts and studs shall have a nominal diameter not less than 12 mm except where manufactured of high strength material in which event the minimum nominal diameter should be 8 mm. Bolts smaller than these sizes or larger than 50 mm nominal diameter require special tightening techniques and should be avoided where possible.

Where bolting is to be unscrewed frequently or where only one or two bolts hold the joint together, an increase in bolting size is recommended.

NOTE: Bolt data is to be obtained from the relevant bolt Standard. The core area (cross-sectional area at root of thread) for bolts to AS/NZS 1110.1 can be calculated from the following equation:

$$\frac{\pi}{4}(D_b - 1.23 P_{th})^2$$

where

P_{th} = bolt thread pitch, in millimetres

D_b = bolt outside diameter, in millimetres

For common bolt sizes, this gives:

Metric bolts	
Size × pitch mm	Root area mm^2
M8 × 1.25	32.8
M10 × 1.5	52.3
M12 × 1.75	76
M16 × 2	144
M20 × 2.5	225
M22 × 2.5	282
M24 × 3	324
M27 × 3	427
M30 × 3.5	519
M36 × 4	759
M42 × 4.5	1045
M48 × 5	1377
M56 × 5.5	1905
M64 × 6	2520
M72 × 6	3282
M80 × 6	4144
M90 × 6	5364
M100 × 6	6740

UNC/UN8 series bolts	
Size inches	Root area mm^2
1/2	81
5/8	130
3/4	195
7/8	270
1	356
1 1/8	470
1 1/4	599
1 3/8	745
1 1/2	907
1 5/8	1084
1 3/4	1278
1 7/8	1487
2	1711
2 1/4	2209
2 1/2	2469
2 3/4	3393
3	4080
3 1/4	4831
3 1/2	5645
3 3/4	6521
4	7462

For other bolts with metric threads, refer to AS 1275 or AS 1721. For bolts with other threads, refer to the appropriate Standard.

3.21.5.4.7 *Stud attachment*

Where tapped holes are provided for studs, and the like, the threads shall be full and clean and shall engage the stud for a length no less than the larger of d_s and—

$$0.75d_s \times \frac{\text{design strength of stud material at design temperature}}{\text{design strength of tapped material at design temperature}}$$

where d_s is the diameter of the stud, except that the thread engagement need not exceed $1.5d_s$.

See Clause 3.19.6.4 for studs in blind holes.

3.21.5.5 *Gaskets*

Gaskets shall be made of materials that are not seriously affected by the contained fluid under all anticipated operating conditions. Non-metallic gaskets shall not be used above 400°C calculation temperature, and shall not be used in non-confining (flat or raised face) flanged joints for the following conditions, except where specifically recommended by the gasket manufacturer—

(a) with toxic or flammable material at or above 8.3 MPa pressure at ambient temperature or equivalent flange rating; and

(b) with non-toxic and non-flammable material at or above 12.5 MPa pressure at ambient temperature or equivalent flange rating.

Gasket materials should be used within the service conditions recommended by the gasket manufacturer.

Self-equalizing types of gasket (O-ring, delta-ring and lens types) are frequently used for high-pressure application. Joints of this kind do not require mechanical loading for gasket seating and, since the gasket reaction can be considered negligible, the total bolting is only that necessary to retain the hydrostatic end-force and any external loads.

3.21.6 Narrow-face flanges with ring-type gaskets

3.21.6.1 *General*

The flange design methods outlined in this Clause (3.21.6) together with Clauses 3.21.1 to 3.21.5 are applicable to circular flanges under internal pressure, with or without external loads on the whole flange assembly, with gaskets that are entirely within the circle enclosed by the bolt holes and with no contact outside this circle. Modifications are given in Clauses 3.21.7 and 3.21.8 for the design of split and non-circular flanges and in Clause 3.21.9 for flanges subject to external pressure.

The method includes provision for seating additional gasket material inside the peripheral gasket, as found on multi-pass heat exchange girth flanges (referred to as pass partition gaskets). Where no such material is present the pass partition terms are not applicable.

Where external loads are to be included in the design, two approaches are presented in this Clause (3.21.6). Firstly, the designer may add an equivalent pressure (due to the external loads) to the calculation pressure, and use this combined pressure to design the flange, or to compare with the pressure rating of a standard flange. This approach provides simple assessment of standard flanges, but is conservative for high pressure flanges.

Secondly, the designer may add the total bolt load (due to the external axial tension and the equivalent axial tension due to the resultant moment on the flange) to the calculated values of W_{m1} and W_{m2}.

The gasket seating stress (y) shall be achieved prior to application of pressure, and any loads that can be applied to the flange between initial bolt up and the commencement of operation (due to weight, piping, thermal loads etc.) shall be included in the determination of the gasket seating bolt load (W_{m2}). Where doubt exists as to the most appropriate external load acting prior to pressurization, the designer shall select the most conservative loading.

3.21.6.2 *Notation*

The notation described below is used in the equations for the design of flanges (see also Figures 3.21.6.2 and 3.21.12.2). Additional and modified notation for flanges with full-face gaskets and reverse flanges is given in Clauses 3.21.11.2 and 3.21.12.2, respectively.

A = outside diameter of flange, or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots, in millimetres

A_b = actual cross sectional area of bolts at root of thread or section of least diameter under stress, in square millimetres.

A_m = total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}, in square millimetres

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions, in square millimetres

$$= \frac{W_{ml}}{S_b}$$

A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating, in square millimetres

= $\frac{W_{m2}}{S_a}$

A_p = total pass partition area of gasket that lies within the gasket ID, in millimetres.

B = inside diameter of flange, in millimetres; when B is less than $20g_1$, it will be optional for the designer to substitute B_1 for B in the equation for longitudinal stress S_H

B_1 = $B + g_1$ for loose-type hub flanges and also for integral-type flanges when f is less than 1

= $B + g_0$ for integral-type flanges when f is equal to or greater than 1

b = effective peripheral gasket or joint-contact-surface seating width, in millimetres (see Note, Clause 3.21.6.4.1(a))

$2b$ = effective peripheral gasket or joint-contact-surface pressure width, in millimetres (see Note, Clause 3.21.6.4.1(a))

b_o = basic gasket seating width, in millimetres (from Table 3.21.6.4(B))

b_p = effective pass partition gasket or joint-contact-surface seating width, in millimetres

$2b_p$ = effective pass partition gasket or joint-contact-surface pressure width, in millimetres

C = bolt circle diameter, in millimetres

c = basic dimension used for the minimum sizing of welds, equal to t_n or t_x, whichever is less, in millimetres

D = diameter of bolt hole, in millimetres

D_b = bolt outside diameter, in millimetres

d = factor, in millimetres to the 3rd power

for integral-type flanges = $\frac{U}{V} h_0 g_0^2$

for loose-type flanges = $\frac{U}{V_L} h_0 g_0^2$

E = modulus of elasticity of flange material at operating temperature (see Table B3), in megapascals

e = factor, in millimetres to the power of minus 1

for integral flanges = F/h_0

for loose-type flanges = F_L/h_0

F = factor for integral-type flanges (from Figure 3.21.6.6(B))

F_{eg} = external axial tensile load on flange (ignoring compressive loads) for gasket seating, in newtons

F_{eo} = external axial tensile load on flange (ignoring compressive loads) for operating condition, in newtons

F_L = factor for loose-type flanges (from Figure 3.21.6.6(D))

f = hub stress correction factor for integral flanges from Figure 3.21.6.6(F) (when greater than 1, this is the ratio of the stress in the small end of hub to the stress in the large end) (for values below limit of figure use $f = 1$)

G = diameter at location of gasket-force, in millimetres; except as noted in Figure 3.21.6.2(a) it is defined as follows:

For flanges covered by this Clause (3.21.6) (see Table 3.21.6.4(B))—

when $b_o \leq 6$ mm, G = mean diameter of gasket contact-face

when $b_o > 6$ mm, G = outside diameter of gasket contact-face minus $2b$

g_0 = thickness of hub at small end, in millimetres

g_1 = thickness of hub at back of flange, in millimetres

H = total hydrostatic end-force, in newtons

= $0.785G^2P$

H_D = hydrostatic end-force on area inside of flange, in newtons

= $0.785B^2P$

H_G = for flanges covered by this Clause (3.21.6), gasket-force (difference between flange design bolt-force and total hydrostatic end-force), in newtons

= $W - H$

H_p = total joint-contact surface compression force, in newtons

= $(2b\pi GmP)+(2b_pL_pm_pP)$

H_T = difference between total hydrostatic end-force and the hydrostatic end-force on area inside of flange, in newtons

= $H - H_D$

h = hub length, in millimetres

h_D = radial distance from the bolt circle to the circle on which H_D acts, as described in Table 3.21.6.5, in millimetres

h_G = radial distance from gasket-force reaction to the bolt circle as described in Table 3.21.6.5, in millimetres

h_0 = factor = $\sqrt{(Bg_0)}$

h_T = radial distance from the bolt circle to the circle on which H_T acts as described in Table 3.21.6.5, in millimetres

J_r = joint rigidity index

K = ratio of outside diameter of flange to inside diameter of flange

= A/B

L = factor = $\dfrac{te+1}{T} + \dfrac{t^3}{d}$

L_p = length of pass partition gasket, in millimetres

M_D = component of moment due to H_D, in newton millimetres

= H_Dh_D

M_{eg} = resultant external moment acting on flange for gasket seating, in newton millimetres

M_{eo} = resultant external moment acting on flange for operating condition, in newton millimetres

M_G = component of moment due to H_G, in newton millimetres

= $H_G h_G$

M_o = total moment (including external loads) acting upon the flange, for operating conditions or gasket seating as may apply, in newton millimetres (see Clause 3.21.6.5). This notation applies to flanges covered by Clauses 3.21.6 to 3.21.9 inclusive and Clause 3.21.12.

M_T = component of moment due to H_T, in newton millimetres

= $H_T h_T$

m = peripheral gasket factor, obtained from Table 3.21.6.4(A) (see Note, Clause 3.21.6.4.1(a))

m_p = pass partition gasket factor, obtained from Table 3.21.6.4(A) (see Note, Clause 3.21.6.4.1(a))

N = width used to determine the basic gasket seating-width, b_o, based upon the possible contact width of the gasket (see Table 3.21.6.4(B)), in millimetres

n = number of bolts

P = calculation pressure, in megapascals. For flanges subject to external pressure see Clause 3.21.9

P_e = equivalent pressure due to external loading, in megapascals

P_b = centreline-to-centreline bolt spacing, in millimetres

P_t = $P + P_e$

P_{th} = Bolt thread pitch, in millimeters

R_{eb} = Specified minimum yield strength of bolt material at ambient conditions, in megapascals

S_a = Design strength for bolt at atmospheric temperature, as given in Table B2, and limited to R_m/3.5, in megapascals

S_b = Design strength for bolt at design temperature, as given in Table B2, and limited R_m/3.5, in megapascals

S_f = design strength for material of flange at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (given in Clause 3.3.1 as f), in megapascals (see Note 1)

S_H = calculated longitudinal stress in hub, in megapascals

S_n = design strength for material of nozzle neck, vessel or pipe wall, at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (given in Clause 3.3.1 as f), in megapascals (see Note 1)

S_R = calculated radial stress in flange, in megapascals

S_T = calculated tangential stress in flange, in megapascals

T = factor involving K (from Figure 3.21.6.6(A))

t = flange thickness, in millimetres

t_n = nominal thickness of shell or nozzle wall to which flange or lap is attached, minus corrosion allowance, in millimetres

t_x = two times the thickness g_0, when the design is calculated as an integral flange, or two times the thickness of shell or nozzle wall required for internal pressure, when the design is calculated as a loose flange, but not less than 6 mm, in millimetres

U = factor involving K (from Figure 3.21.6.6(A))

V = factor for integral-type flanges (from Figure 3.21.6.6(C))

V_L = factor for loose-type flanges (from Figure 3.21.6.6(E))

W = flange design bolt-force, for the operating conditions or gasket seating, as may apply, in newtons (see Clause 3.21.6.4.4 for flanges covered by this Clause (3.21.6)

W_{m1} = minimum required bolt-force for operating conditions (see Clause 3.21.6.4), in newtons

For flange pairs used to contain a tube sheet for a floating head for a U-tube type of heat exchanger, or for any other similar design, W_{m1} shall be the larger of the values as individually calculated for each flange, and that value shall be used for both flanges.

W_{m2} = minimum required bolt-force for gasket seating (see Clause 3.21.6.4), in newtons

w = width used to determine the basic gasket seating width b_o, based upon the contact width between the flange facing and the gasket (see Table 3.21.6.4(B)), in millimetres

Y = factor involving K (from Figure 3.21.6.6(A))

y = peripheral gasket or joint-contact-surface seating stress (see Note in Clause 3.21.6.4.1), in megapascals

y_p = pass partition gasket or joint-contact-surface seating stress (see Note in Clause 3.21.6.4.1); in megapascals

Z = factors involving K (from Figure 3.21.6.6(A))

3.21.6.3 *Circular flange types*

For purposes of calculation, there are three types:

(a) *Loose-type flanges* This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall, and designs where the method of attachment is not considered to give the mechanical strength equivalent of integral attachment. See Figure 3.21.3(f), (g), (h), (j), (k) and (m) for typical loose-type flanges and Figure 3.21.6.2 for location of the forces and moments.

(b) *Integral-type flanges* This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel or pipe wall, butt-welded thereto, or attached by other forms of arc or gas welding of such a nature that the flange and nozzle neck, vessel, or pipe wall is considered to be the equivalent of an integral structure. In welded construction, the nozzle neck, vessel, or pipe wall is considered to act as a hub. See Figure 3.21.3(a), (c) and (l), for typical integral-type flanges and Figure 3.21.6.2 for location of the forces and moments.

(c) *Optional-type flanges* This type covers designs where the attachment of the flange to the nozzle neck, vessel, or pipe wall is such that the assembly is considered to act as a unit, which shall be calculated as an integral flange, except that for simplicity the designer may calculate the fabrication as a loose-type flange provided that none of the following values is exceeded:

$g_0 = 16\,\text{mm}; \dfrac{B}{g_0} = 300; P = 2.1\,\text{MPa};$

design temperature = 370°C

See Figure 3.21.3(b), (d), (e), (n) and (p) for typical optional-type flanges.

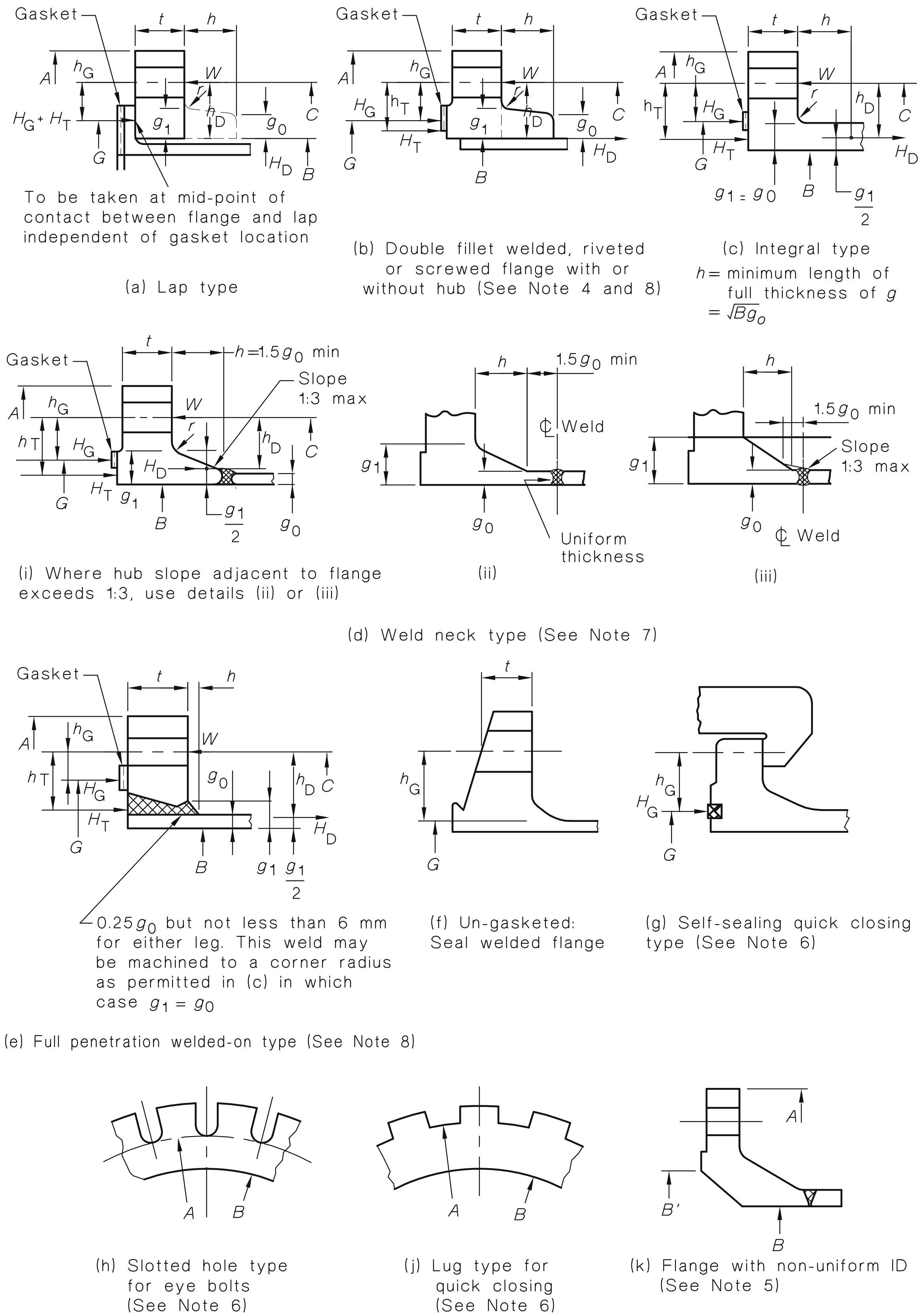


FIGURE 3.21.6.2 FLANGE NOTATION

NOTES TO FIGURE 3.21.6.2:

1 For hub tapers 6° or less, use $g_0 = t_n$.

2 Fillet radius r to be at least $0.25g_1$ but not less than 5 mm.

3 Raised, tongue-groove, male and female, and ring joint facings shall be in excess of the required minimum flange thickness t.

4 The vessel or pipe wall shall not be considered to have any value as a hub for this attachment or where the back weld is a fillet weld only.

5 In calculation, use $K = A/B'$ in place of $K = A/B$.

A1 6 Closure elements shall comply with Items (h) to (k) of Clause 3.27.2.

7 In details (d)(i) to (d)(iii), h is measured to the intersection of the hub taper and the outside surface of the shell or nozzle; this may or may not include the weld metal.

8 Loads and dimensions shown in Item (b) apply for calculation as a loose type flange. Item (e) applies for calculation as an integral type flange.

3.21.6.4 *Bolt-forces*

3.21.6.4.1 *Bolt-forces for non-self-energizing type gaskets*

A2 (a) *Force for operating conditions* The required bolt-force for the operating conditions, W_{m1}, shall be sufficient to resist the following, all at the design temperature:

(i) The hydrostatic end-force H, exerted by the calculation pressure on the area bounded by the diameter of gasket reaction.

(ii) The calculated equivalent additional bolt load due to external loading.

(iii) A compression-force, H_p, on the gasket or joint-contact-surface that experience has shown to be sufficient to ensure a tight joint.

NOTE: Tables 3.21.6.4(A) and 3.21.6.4(B) list some commonly used gasket materials and contact facings, with suggested values of m, b, and y that have proved satisfactory in actual service. Alternative values may be obtained by testing to ASTM F586 *Test method for leak rates versus y stresses and m factors for gaskets* (withdrawn), or an equivalent National Standard. Values that are too low may result in leakage at the joint, without affecting the safety of the design. The primary proof that the values are adequate is the hydrostatic test.

Where flanges are subject to external loads or moments, these are converted to their pressure equivalents, which are then added to the internal pressure (P) to give an equivalent pressure (P_e) in accord with Equation 3.21.6.4.1(1):

$$P_e = P + \frac{4F_{eo}}{\pi G^2} + \frac{16M_{eo}}{\pi G^3} \quad \text{. . . 3.21.6.4.1(1)}$$

The required bolt-force for the operating conditions, W_{m1}, shall be determined using either Equation 3.21.6.4.1(2) or Equation 3.21.6.4.1(3):

$$W_{m1} = \ = H + H_p$$

$$= 0.785\ G^2 P_e + 2b\pi GmP_e + 2b_p L_p m_p P_e \quad \text{. . . 3.21.6.4.1(2)}$$

or

$$= 0.785\ G^2 P + 2b\pi GmP + 2b_p L_p m_p P + F_{eo} + \frac{4M_{eo}}{G} \quad \text{. . . 3.21.6.4.1(3)}$$

(b) *Gasket seating-force* Before a tight joint can be obtained it is necessary to seat the gasket or joint-contact-surface by applying a minimum initial gasket seating force (under atmospheric conditions without the presence of internal pressure) determined in accordance with Equation 3.21.6.4.1(4)—

$$W_{m2} = \pi bGy + b_p L_p y_p + F_{eg} + \frac{4M_{eg}}{G} \quad \text{. . . 3.21.6.4.1(4)}$$

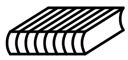
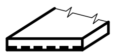
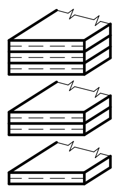
A2 For flange pairs that contain two gaskets, (e.g. the fixed tube sheet for a shell and tube heat exchanger), or for other similar design, and where the operating pressure, flanges or gaskets (or some combination of those factors) are not the same, W_{m1} and W_{m2} shall be the larger of the values obtained from either Equation 3.21.6.4.1(2) or Equation 3.21.6.4.1(3) and Equation 3.21.6.4.1(4), respectively, as individually calculated for each flange and gasket, and the most severe value shall be used for both flanges.

The need for providing sufficient bolt-force to seat the gasket or joint-contact-surfaces in accordance with Equation 3.21.6.4.1(4) will prevail on many low-pressure designs and with facings and materials that require a high seating-force and where the bolt-force calculated by Equations 3.21.6.4.1(1) for the operating conditions is insufficient to seal the joint. Accordingly, it is necessary to furnish bolting and to pre-tighten the bolts to provide a bolt-force sufficient to satisfy both of these requirements, each one being individually investigated. When Equation 3.21.6.4.1(2) governs, flange proportions will be a function of the bolting instead of internal pressure.

In practice flanges are generally tightened to a bolt tension greater than that calculated above to ensure a tight joint under both operating and hydrotest conditions (for further information, see ASME BPV VIII-1 Appendix S and ASME PCC-1 Appendix O). Tensions achieved in industry are typically in the range of 35% to 70% of bolt yield strength (R_e).

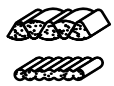
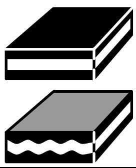
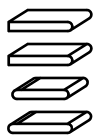
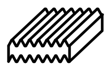
TABLE 3.21.6.4(A)

GASKET MATERIALS AND CONTACT FACINGS

Gasket factor (*m*) for operating conditions and minimum design seating strength (*y*) (see Note 1)					Refer to Table 3.21.6.4(B)	
Gasket material (thickness or composition as noted) (see Note 2)		**Gasket factor (*m*)**	**Min. design seating stress (*y*) MPa**	**Sketches and notes**	**Use facing sketch**	**Use column**
Self-energizing types: O-rings, metallic, elastomer, other gasket types considered as self-sealing		0	0	—	—	—
Elastomers without fabric or a high percent of asbestos fibre Below 75 Shore Durometer 75 or higher Shore Durometer		0.50 1.00	0 1.4		1 (a, b, c, d), 4, 5	
Vegetable fibre		1.75	7.6			
Spiral-wound metal, graphite or PTFE filled	Carbon Stainless or monel	2.50 3.00	69.0 69.0		1 (a, b)	
CNAF with a suitable binder (glass, aramid, carbon fibre or any combination) for the operating conditions	1.0 mm 1.5 mm 3.0 mm	3.5 3.7 4.5	25 25 25			
Restructured and reinforced PTFE Low compressibility (<5%)	1.0 mm 2.0 mm 3.0 mm	4.7 5 5.2	10 10 10			
Restructured and reinforced PTFE Medium compressibility (5-10%)	All	5	10		1 (a, b, c, d), 4, 5	II
Restructured and reinforced PTFE High compressibility (15-30%)	All	2.8	5			
Expanded PTFE (compressibility > 50%)	1.5 3.0	2.5 3.2	20 25			
Elastomers with cotton fabric insertion		1.25	2.8			
Elastomers with asbestos fabric insertion, with or without wire reinforcement (see Note 2)	3-ply 2-ply 1-ply	2.25 2.50 2.75	15.2 20.0 25.5			

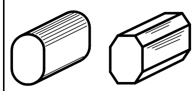
(*continued*)

TABLE 3.21.6.4(A) (*continued*)

Gasket factor (*m*) for operating conditions and minimum design seating strength (*y*) (see Note 1)					Refer to Table 3.21.6.4(B)	
Gasket material (thickness or composition as noted) (see Note 2)		Gasket factor (*m*)	Min. design seating stress (*y*) MPa	Sketches and notes	Use facing sketch	Use column
Corrugated metal, graphite inserted or *Corrugated metal, jacketed graphite filled*	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.50 2.75 3.00 3.25 350	20.0 25.5 31.0 38.0 45.0		1 (a,b)	II
Corrugated metal	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	2.75 3.00 3.25 3.50 3.75	25.5 31.0 38.0 45.0 52.5		1 (a, b, c, d)	
Exfoliated graphite or PTFE laminate facing on solid metal core	All metal cores & core profiles	2.0	17.2		1 (a,b)	
Flat metal jacketed graphite filled	Soft aluminium Soft copper or brass Iron or soft steel Monel 4-6% chrome Stainless steels	3.25 3.50 3.75 3.50 3.75 3.75	38.0 45.0 52.5 55.5 62.0 62.0		1 (a, b, c, d), 2. For 1 (c, d), 2, see Note 3	
Grooved metal	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4-6% chrome Stainless steels	3.25 3.50 3.75 3.75 4.25	38.0 45.0 52.5 62.0 70.0		1 (a, b, c, d), 2,3	I

(*continued*)

TABLE 3.21.6.4(A) (*continued*)

Gasket factor (*m*) for operating conditions and minimum design seating strength (*y*) (see Note 1)					Refer to Table 3.21.6.4(B)	
Gasket material (thickness or composition as noted) (see Note 2)		**Gasket factor (m)**	**Min. design seating stress (y) MPa**	**Sketches and notes**	**Use facing sketch**	**Use column**
Soft flat metal	Soft aluminium	4.00	61.0		1 (a, b, c, d), 2, 3, 4, 5	I
	Soft copper or brass	4.75	90.0			
	Iron or soft steel	5.50	124.0			
	Monel or 4-6% chrome	6.00	151.0			
	Stainless steels	6.50	180.0			
Ring joint	Iron or soft steels	5.50	124.0		6	
	Monel or 4-6% chrome	6.00	151.0			
	Stainless steels	6.50	180.0			

NOTES:

1 This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table 3.21.6.4(B). The design values and other details given in this table are suggested only and are not mandatory.

2 Asbestos-type gaskets are generally not permitted to be used. The values listed are only for reference for previous designs.

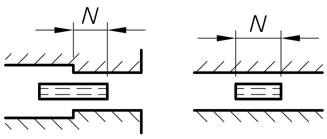
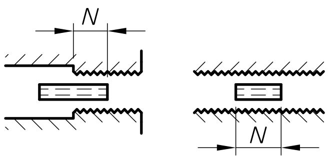
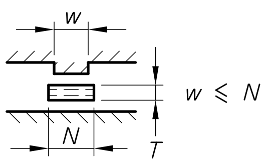
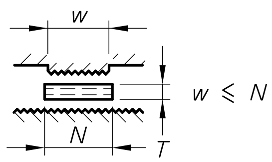
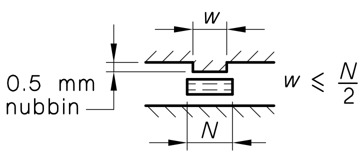
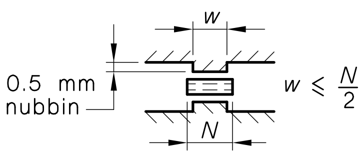
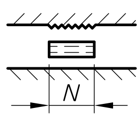
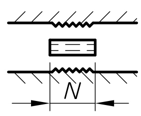
CNAF indicates compressed non-asbestos fibre.

3 The surface of a gasket having a lap should not be against the nubbin.

4 The values given in this Table for asbestos-based gaskets may not apply for gaskets constructed of non-asbestos fibre substitutes (e.g. aramid, carbon/aramid, glass/aramid). The gasket manufacturer should be consulted for guidance on the gasket choice, gasket factors, seating stress, chemical and thermal resistance and contact facing. If special bolt tightening procedures are required, these shall be communicated to the purchaser.

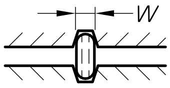
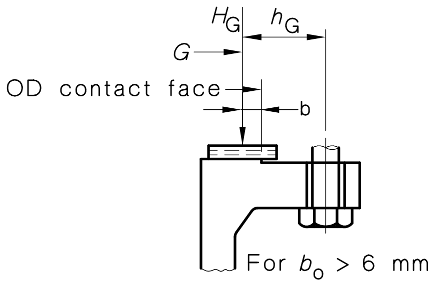
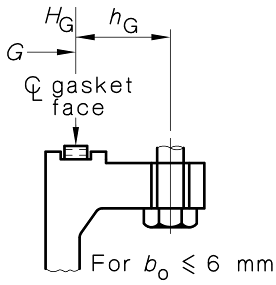
TABLE 3.21.6.4(B)

EFFECTIVE GASKET WIDTH (See Clause 3.21.4.3)

Facing sketch (exaggerated)	Basic gasket seating width (b_o)	
	Column I†	Column II†
1(a) 	$\frac{N}{2}$	$\frac{N}{2}$
1(b)* 		
1(c) 	$\frac{w+T}{2}$ but not to exceed $\frac{w+N}{4}$	$\frac{w+T}{2}$ but not to exceed $\frac{w+N}{4}$
1(d)* 		
2 	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
3 	$\frac{N}{4}$	$\frac{3N}{8}$
4* 	$\frac{3N}{8}$	$\frac{7N}{16}$
5* 	$\frac{N}{4}$	$\frac{3N}{8}$

(*continued*)

TABLE 3.21.6.4(B) (*continued*)

Facing sketch (exaggerated)	Basic gasket seating width (b_o)	
	Column I†	Column II†
6 	$\frac{w}{8}$	
EFFECTIVE GASKET SEATING WIDTH (b)		
$b = 2.52\sqrt{b_o}$, when $b_o > 6$ mm		$b = b_o$, when $b_o \leq 6$ mm
LOCATION OF GASKET FORCE REACTION		
		NOTE: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

* Where serrations do not exceed 0.5 mm width spacing, sketches 1 (b) and 1 (d) shall be used.

† See Table 3.21.6.4(A).

NOTE: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

3.21.6.4.2 *Bolt-forces for self-energizing type gaskets*

Bolt-forces for self-energizing type gaskets shall comply with the following:

(a) *Force for operating conditions* The required bolt-force for the operating conditions W_m, shall be sufficient to resist the hydrostatic end-force H, exerted by the calculation pressure on the area bounded by the outside diameter of the gasket. H_p is to be considered as zero (0) for all self-energizing gaskets except that certain seal configurations which generate axial forces shall be considered.

(b) *Gasket seating-forces* Self-energizing gaskets may be considered to require an inconsequential amount of bolting-force to produce a seal, i.e. $W_{m2} = 0$. Bolting, however, shall be pretightened to provide a bolt-force sufficient to withstand the hydrostatic end-force H.

3.21.6.4.3 *Total required and actual bolt areas A_m and A_b*

The total cross-sectional area of bolts A_m, required for both the operating conditions and gasket seating is the greater of the values for A_{m1} and A_{m2} where $A_{m1} = W_{m1}/S_b$ and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolts A_b, will not be less than A_m.

3.21.6.4.4 *Flange design bolt-forces W*

The bolt-forces used in the design of the flange shall be the values obtained from Equations 3.21.6.4.4(1) and 3.21.6.4.4(2):

For operating conditions—

$$W = W_{m1} \qquad \text{. . . 3.21.6.4.4(1)}$$

For gasket seating—

$$W = \frac{(A_m + A_b)S_a}{2} \qquad \text{. . . 3.21.6.4.4(2)}$$

In addition to the minimum requirements for safety, Equation 3.21.6.4.4(2) provides a margin against abuse of the flange from overbolting. Since the margin against such abuse is needed primarily for the initial bolting-up operation which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions.

3.21.6.5 *Flange moments*

In the calculation of flange stresses, the moment of force acting on the flange is the product of the force and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the force producing the moment (see Figure 3.21.6.2). No credit shall be taken for the reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof.

For the operating conditions, the total flange moment M_o, is the sum of the three individual moments M_D, M_T, and M_G, as defined in Clause 3.21.6.2 and based on the flange design bolt-force of Equation 3.21.6.4.4(1) with moment arms as given in Table 3.21.6.5.

For gasket seating, the total flange moment M_o, is based on the flange design bolt-force of Equation 3.21.6.4.4(2), which is opposed only by the gasket force in which case—

$$M_o = Wh_G \qquad \text{. . . 3.21.6.5}$$

TABLE 3.21.6.5

MOMENT ARMS FOR FLANGE FORCES UNDER OPERATING CONDITIONS

Type of flange	Values		
	h_D	h_T	h_G
Integral-type flanges (see Figure 3.21.3(a), (c) and (l))	$\frac{C-B-g_1}{2}$	$\frac{C-B}{4}+\frac{h_G}{2}$	$\frac{C-G}{2}$
Loose-type; except lap-joint flanges (see Figure 3.21.3(f), (g), (h), (j) and (m)), and optional-type flanges (see Figure 3.21.3(b), (d) and (e)).	$\frac{C-B}{2}$	$\frac{h_D+h_G}{2}$	$\frac{C-G}{2}$
Lap-joint flanges (see Figure 3.21.3(k))	$\frac{C-B}{2}$	$\frac{C-G}{2}$	$\frac{C-G}{2}$
Reverse-type flange (see Figure 3.21.12.2(a))	$\frac{C-B+g_1-2g_0}{2}$	$0.5\left(C-\frac{B+G}{2}\right)$	$\frac{C-G}{2}$

3.21.6.6 *Calculation of flange stresses*

The stresses in the flange shall be determined for both the operating conditions and gasket seating, whichever controls, in accordance with the following:

(a) *For integral-type flanges and all hub-type flanges*:

Longitudinal hub stress—

$$S_H = \frac{fM_o}{Lg_1^2 B} \qquad \text{. . . 3.21.6.6(1)}$$

Radial flange stress—

$$S_R = \frac{(1.33te + 1)M_o}{Lt^2 B} \qquad \text{. . . 3.21.6.6(2)}$$

Tangential flange stress—

$$S_T = \frac{YM_o}{t^2 B} - ZS_R \qquad \text{. . . 3.21.6.6(3)}$$

(b) *For loose-type ring flanges (including optional-type calculated as loose-type) having a rectangular cross-section*:

Tangential flange stress—

$$S_T = \frac{YM_o}{t^2 B}$$

and $S_R = 0$; $S_H = 0$

NOTE: See Figure 3.21.6.6(A) for values of Y and Z. See Table 3.21.6.6 for flange factors in equation form.

3.21.6.7 *Flange design strengths*

The flange stresses calculated by the equations in Clause 3.21.6.6 shall not exceed any of the following values:

(a) Longitudinal hub stress S_H not greater than S_f for cast iron (see Note) and, except as otherwise limited by (b) and (c), not greater than $1.5S_f$ for materials other than cast iron.

NOTE: When the flange material is cast iron, particular care should be taken when tightening the bolts to avoid excessive stress that may break the flange; an attempt should be made to apply no greater wrenching effort than is needed to ensure tightness in the hydrostatic test.

(b) Longitudinal hub stress S_H not greater than the smaller of $1.5S_f$ and $1.5S_n$ for optional-type flanges designed as integral (Figure 3.21.3(b), (d) and (e)), also integral type (Figure 3.21.3(c)) where the neck material constitutes the hub of the flange.

(c) Longitudinal hub stress S_H not greater than the smaller of $1.5S_f$ and $2.5S_n$ for integral-type flanges with hub welded to the neck, pipe or vessel wall (Figure 3.21.3 (a)).

(d) Radial flange stress S_R no greater than S_f.

(e) Tangential flange stress S_T no greater than S_f.

(f) Also $\frac{S_H + S_R}{2}$ no greater than S_f and

$\frac{S_H + S_T}{2}$ no greater than S_f.

For hub flanges attached as shown in Figure 3.21.3(f), (g), (h), (j) and (m), the nozzle neck, vessel or pipe wall shall be considered to have no value as a hub.

In the case of loose-type flanges with laps, as shown in Figure 3.21.3(k), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed $0.8S_n$ for the material of the lap, as defined in Clause 3.21.6.2. In the case of welded flanges shown in Figure 3.21.3(b), (c), (d), (e), (f), (g), (h) and (j), where the nozzle neck, vessel or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed $0.8S_n$. The shearing stress shall be calculated on the basis of W_{m1} or W_{m2}, as defined in Clause 3.21.6.2, whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

3.21.6.8 *Flange rigidity*

To ensure sufficient rigidity to prevent leakage, flanges that are greater than DN600, or that comply with a European flange Standard (i.e. with a safety factor less than 3.5 against ultimate strength) shall satisfy the requirements listed below, under both gasket seating and operating conditions:

(a) $174\dfrac{M_0 V}{L E g_0^2 h_0} < 1.0$, for integral and optional type flanges.

(b) $261\dfrac{M_0 V}{L E g_0^2 h_0} < 1.0$, for loose type flanges with hubs.

(c) $547\dfrac{M_0}{E t^3 \ln(K)} < 1.0$, for loose type flanges without hubs

where $K = \dfrac{\text{flange outside diameter}}{\text{flange inside diameter}}$.

In the case of flanges that do not comply with an alternative Standard listed in Clause 3.21.1 and that are smaller than DN600, it is strongly recommended they satisfy the requirements shown above.

Where a flange does not satisfy these rigidity requirements, it shall—

(i) be modified to meet the requirements; or

(ii) be de-rated to a lower pressure/temperature and used appropriately; or

(iii) undergo finite element analysis to more accurately assess whether rigidity is suitable for the application.

The rigidity limits given above are somewhat dependent on gasket type, thickness, and operating temperature. Thicker gaskets in some materials may be more tolerant of flange rotation, and so may tolerate a higher value. Similarly, thinner gaskets may need lower values to control leakage. Non-metallic gaskets operating close to their temperature limit generally require lower limits of flange rotation.

Flange design with self-energizing seals (e.g. 'O' rings) need not satisfy these requirements.

3.21.6.9 *Finite element analysis of flanges*

Where finite element analysis (FEA) is used to evaluate the design of a flanged assembly the analysis shall include the following requirements:

(a) The FEA model shall contain at least 2 elements through the thickness of the shell that the flange is attached to, and the flange itself.

(b) Gaskets shall be modelled with representative or actual properties, including consideration of any non-linear characteristics (where present).

(c) Contact elements shall be used between the gasket and mating flange surfaces to allow the effects of rotation to be included.

(d) Bolts shall be included in the model with the actual expected/design installed tensions. Friction between the nut and flange under the applied bolt tension shall be considered as the installed load is often such that no sliding occurs between the nut and flange, which can result in significant bolt stresses if there are high relative thermal expansions involved.

(e) Different temperatures shall be applied to the different components to allow the effects of different radial and axial expansion to be evaluated.

(f) All loads shall be applied to the respective components including pressure, temperature, weight effects, external applied loads.

(g) Primary stresses (membrane, and membrane + bending) shall be limited to 90% of material yield strength at temperature. Any secondary stresses beyond yield shall be local only. The gasket width that ‘unloads’ (i.e. no residual pressure) under the analysed conditions shall be limited to a maximum of 20% of the actual gasket width.

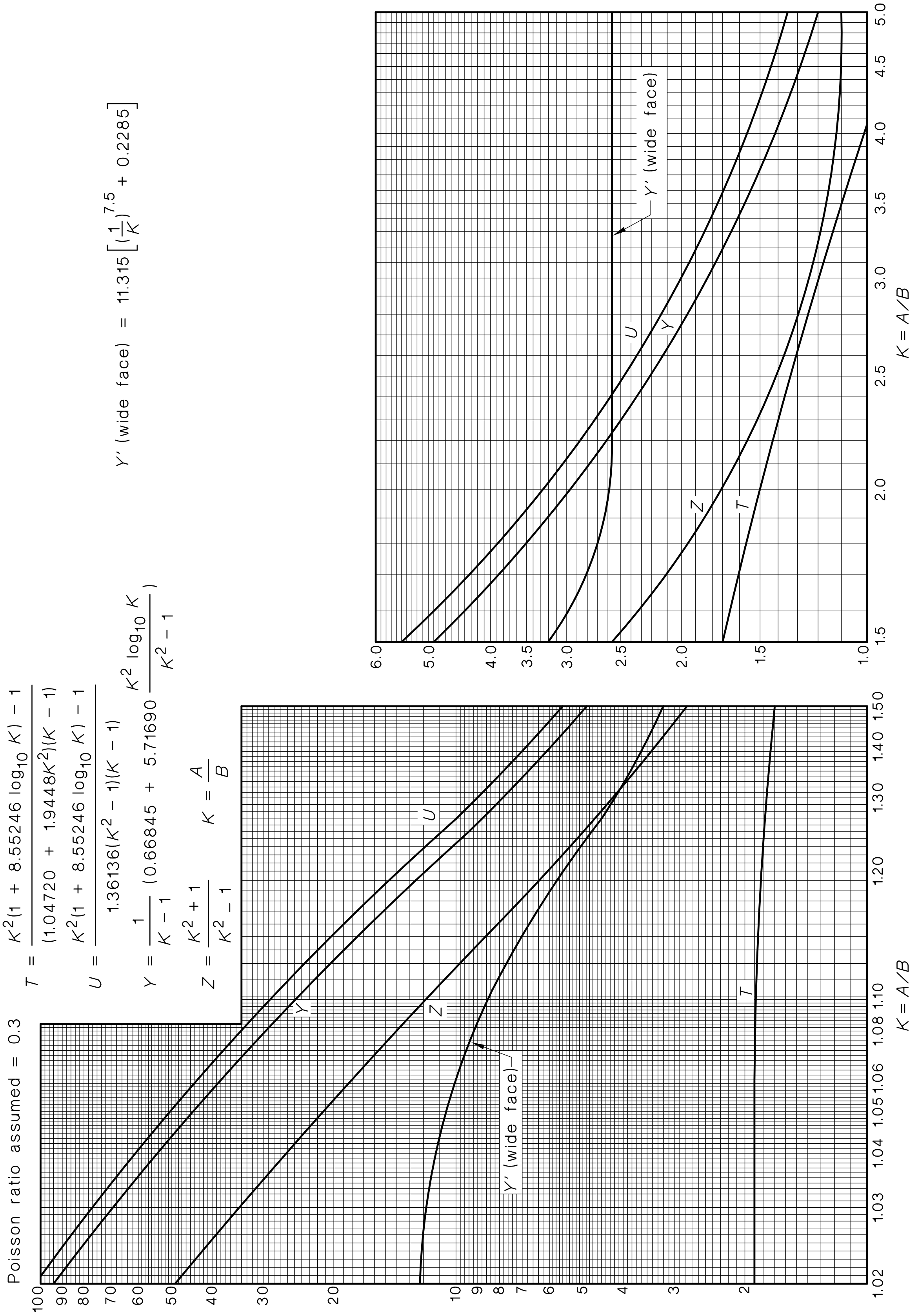


FIGURE 3.21.6.6(A) T, U, Y, Y' AND Z (TERMS INVOLVING K)

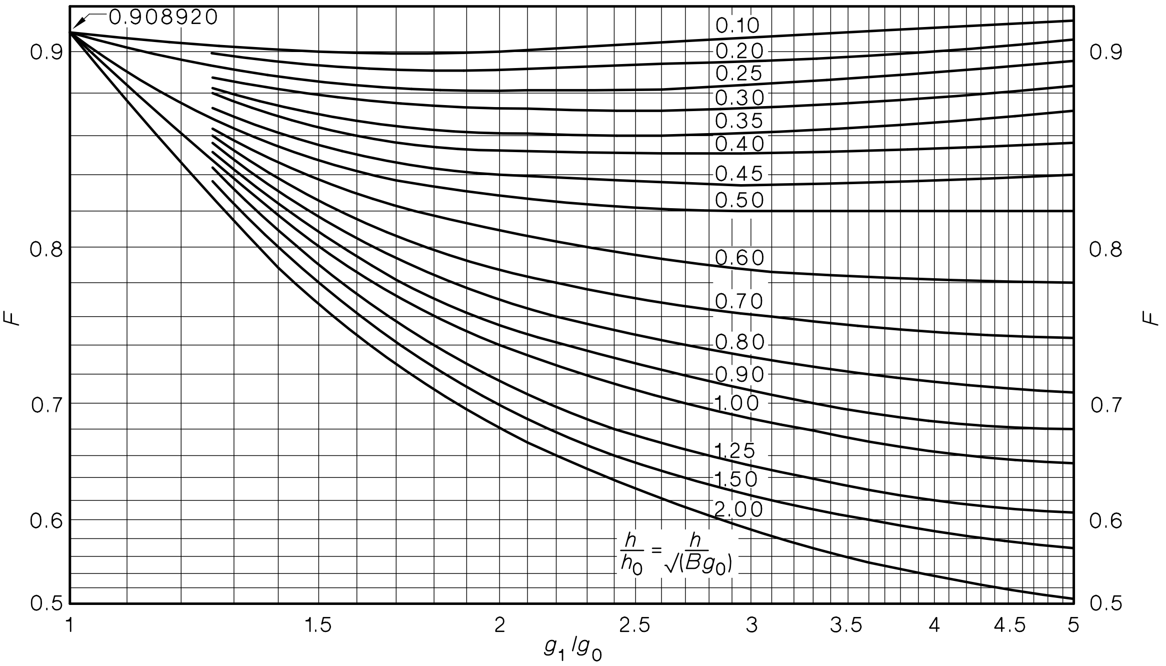


FIGURE 3.21.6.6(B) VALUES OF *F* (INTEGRAL FLANGE FACTORS)

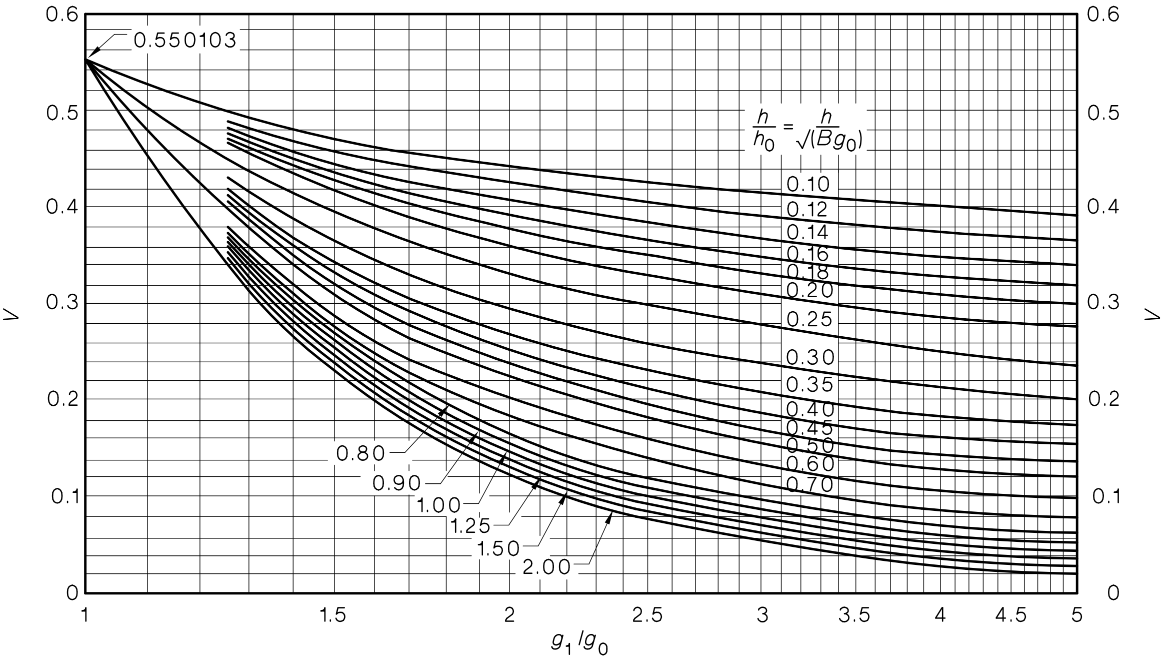


FIGURE 3.21.6.6(C) VALUES OF *V* (INTEGRAL FLANGE FACTORS)

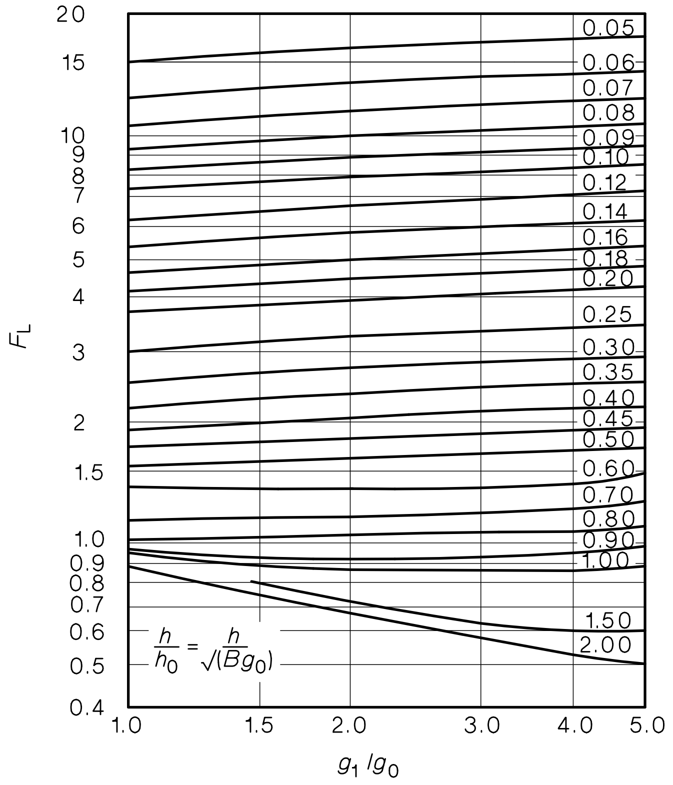


FIGURE 3.21.6.6(D) VALUES OF F_L (LOOSE HUB FLANGE FACTORS)

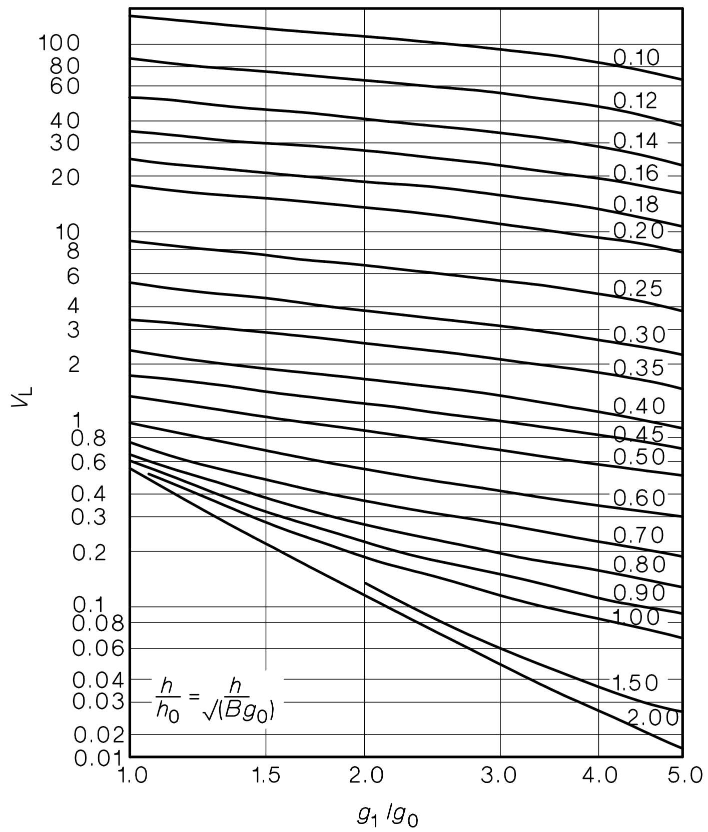


FIGURE 3.21.6.6(E) VALUES OF V_L (LOOSE HUB FLANGE FACTORS)

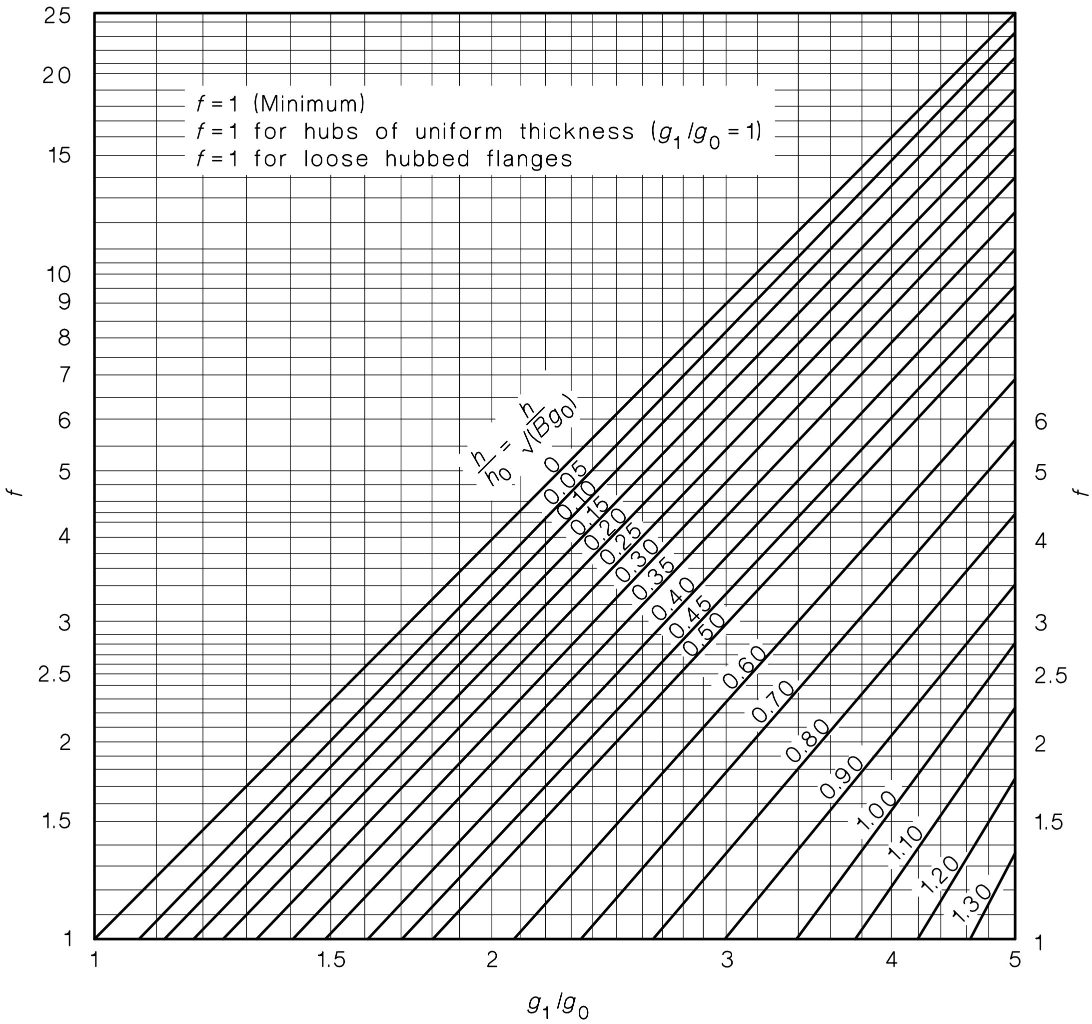


FIGURE 3.21.6.6(F) VALUES OF *f* (HUB STRESS CORRECTION FACTOR)

TABLE 3.21.6.6

FLANGE FACTORS IN EQUATION FORM

Integral flange

Factor F per Figure 3.21.6.6(B) is solved by $F = -\dfrac{E_6}{\left(\dfrac{C}{2.73}\right)^{1/4}\dfrac{(1+A)^3}{C}}$

Factor V per Figure 3.21.6.6(C) is solved by $V = \dfrac{E_4}{\left(\dfrac{2.73}{C}\right)^{1/4}(1+A)^3}$

Factor f per Figure 3.21.6.6(F) is solved by $f = C_{36}/(1+A)$

The values used in the above equations are solved using Equation (1) through (45), below, based on the values g_o, g_1, h, and h_o as defined by Clause 3.21.6.2.

Loose hub flange

Factor F_L per Figure 3.21.6.6(D) is solved by $F_L = -\dfrac{C_{18}\left(\dfrac{1}{2}+\dfrac{A}{6}\right)+C_{21}\left(\dfrac{1}{4}+\dfrac{11A}{84}\right)+C_{24}\left(\dfrac{1}{70}+\dfrac{A}{106}\right)-\left(\dfrac{1}{40}+\dfrac{A}{72}\right)}{\left(\dfrac{C}{2.73}\right)^{1/4}\dfrac{(1+A)^3}{C}}$

Factor V_L per Figure 3.21.6.6(E) is solved by $V_L = \dfrac{\dfrac{1}{4}-\dfrac{C_{24}}{5}-\dfrac{3C_{21}}{2}-C_{18}}{\left(\dfrac{2.73}{C}\right)^{1/4}(1+A)^3}$

Factor f per Figure 3.21.6.6(F) is set equal to 1 $f = 1$

The values used in the above equations are solved using Equations (1) through (5), (7), (9), (10), (12), (14), (16), (18), (20), (23) and (26) below, based on the values of g_1, g_o, h, and h_o as defined by Clause 3.21.6.2.

Equations

(1)	A	=	$(g_1/g_0) - 1$
(2)	C	=	$43.68(h/h_0)^4$
(3)	C_1	=	$1/3 + A/12$
(4)	C_2	=	$5/42 + 17A/336$
(5)	C_3	=	$1/210 + A/360$
(6)	C_4	=	$11/360 + 59A/5040 + (1 + 3A)/C$
(7)	C_5	=	$1/90 + 5A/1008 - (1 + A)^3/C$
(8)	C_6	=	$1/120 + 17A/5040 + 1/C$
(9)	C_7	=	$215/2772 + 51A/1232 + (60/7 + 225A/14 + 75A^2/7 + 5A^3/2)/C$
(10)	C_8	=	$31/6930 + 128A/45\,045 + (6/7 + 15A/7 + 12A^2/7 + 5A^3/11)/C$
(11)	C_9	=	$533/30\,240 + 653A/73\,920 + (1/2 + 33A/14 + 39A^2/28 + 25A^3/84)/C$
(12)	C_{10}	=	$29/3780 + 3A/704 - (1/2 + 33A/14 + 81A^2/28 + 13A^3/12)/C$
(13)	C_{11}	=	$31/6048 + 1763A/665\,280 + (1/2 + 6A/7 + 15A^2/28 + 5A^3/42)/C$
(14)	C_{12}	=	$1/2925 + 71A/300\,300 + (8/35 + 18A/35 + 156A^2/385 + 6A^3/55)/C$
(15)	C_{13}	=	$761/831\,600 + 937A/1\,663\,200 + (1/35 + 6A/35 + 11A^2/70 + 3A^3/70)/C$
(16)	C_{14}	=	$197/415\,800 + 103A/332\,640 - (1/35 + 6A/35 + 17A^2/70 + A^3/10)/C$
(17)	C_{15}	=	$233/831\,600 + 97A/554\,400 + (1/35 + 3A/35 + A^2/14 + 2A^3/105)/C$
(18)	C_{16}	=	$C_1 C_7 C_{12} + C_2 C_8 C_3 + C_3 C_8 C_2 - (C_3^2 C_7 + C_8^2 C_1 + C_2^2 C_{12})$
(19)	C_{17}	=	$[C_4 C_7 C_{12} + C_2 C_8 C_{13} + C_3 C_8 C_9 - (C_{13} C_7 C_3 + C_8^2 C_4 + C_{12} C_2 C_9)]/C_{16}$

(*continued*)

TABLE 3.21.6.6 (*continued*)

			Equations
(20)	C_{18}	=	$[C_5\,C_7\,C_{12} + C_2\,C_8\,C_{14} + C_3\,C_8\,C_{10} - (C_{14}\,C_7\,C_3 + C_8^2\,C_5 + C_{12}\,C_2\,C_{10})]/C_{16}$
(21)	C_{19}	=	$[C_6\,C_7\,C_{12} + C_2\,C_8\,C_{15} + C_3\,C_8\,C_{11} - (C_{15}\,C_7\,C_3 + C_8^2\,C_6 + C_{12}\,C_2\,C_{11})]/C_{16}$
(22)	C_{20}	=	$[C_1\,C_9\,C_{12} + C_4\,C_8\,C_3 + C_3\,C_{13}\,C_2 - (C_3^2\,C_9 + C_{13}\,C_8\,C_1 + C_{12}\,C_4\,C_2)]/C_{16}$
(23)	C_{21}	=	$[C_1\,C_{10}\,C_{12} + C_5\,C_8\,C_3 + C_3\,C_{14}\,C_2 - (C_3^2\,C_{10} + C_{14}\,C_8\,C_1 + C_{12}\,C_5\,C_2)]/C_{16}$
(24)	C_{22}	=	$[C_1\,C_{11}\,C_{12} + C_6\,C_8\,C_3 + C_3\,C_{15}\,C_2 - (C_3^2\,C_{11} + C_{15}\,C_8\,C_1 + C_{12}\,C_6\,C_2)]/C_{16}$
(25)	C_{23}	=	$[C_1\,C_7\,C_{13} + C_2\,C_9\,C_3 + C_4\,C_8\,C_2 - (C_3\,C_7\,C_4 + C_8\,C_9\,C_1 + C_2^2\,C_{13})]/C_{16}$
(26)	C_{24}	=	$[C_1\,C_7\,C_{14} + C_2\,C_{10}\,C_3 + C_5\,C_8\,C_2 - (C_3\,C_7\,C_5 + C_8\,C_{10}\,C_1 + C_2^2\,C_{14})]/C_{16}$
(27)	C_{25}	=	$[C_1\,C_7\,C_{15} + C_2\,C_{11}\,C_3 + C_6\,C_8\,C_2 - (C_3\,C_7\,C_6 + C_8\,C_{11}\,C_1 + C_2^2\,C_{15})]/C_{16}$
(28)	C_{26}	=	$-(C/4)^{1/4}$
(29)	C_{27}	=	$C_{20} - C_{17} - 5/12 - [C_{17}\,(C/4)^{1/4}]$
(30)	C_{28}	=	$C_{22} - C_{19} - 1/12 - [C_{19}\,(C/4)^{1/4}]$
(31)	C_{29}	=	$-(C/4)^{1/2}$
(32)	C_{30}	=	$-(C/4)^{3/4}$
(33)	C_{31}	=	$3\mathrm{A}/2 + C_{17}(C/4)^{3/4}$
(34)	C_{32}	=	$1/2 + C_{19}\,(C/4)^{3/4}$
(35)	C_{33}	=	$0.5C_{26}\,C_{32} + C_{28}\,C_{31}\,C_{29} - (0.5\,C_{30}\,C_{28} + C_{32}\,C_{27}\,C_{29})$
(36)	C_{34}	=	$1/12 + C_{18} - C_{21} + C_{18}(C/4)^{1/4}$
(37)	C_{35}	=	$-C_{18}\,(C/4)^{3/4}$
(38)	C_{36}	=	$(C_{28}\,C_{35}\,C_{29} - C_{32}\,C_{34}\,C_{29})/C_{33}$
(39)	C_{37}	=	$[0.5\,C_{26}\,C_{35} + C_{34}\,C_{31}\,C_{29} - (0.5\,C_{30}\,C_{34} + C_{35}\,C_{27}\,C_{29})]/C_{33}$
(40)	E_1	=	$C_{17}\,C_{36} + C_{18} + C_{19}\,C_{37}$
(41)	E_2	=	$C_{20}\,C_{36} + C_{21} + C_{22}\,C_{37}$
(42)	E_3	=	$C_{23}\,C_{36} + C_{24} + C_{25}\,C_{37}$
(43)	E_4	=	$1/4 + C_{37}/12 + C_{36}/4 - E_3/5 - 3E_2/2 - E_1$
(44)	E_5	=	$E_1(1/2 + A/6) + E_2\,(1/4 + 11A/84) + E_3(1/70 + A/105)$
(45)	E_6	=	$E_5 - C_{36}\,(7/120 + A/36 + 3A/C) - 1/40 - A/72 - C_{37}(1/60 + A/120 + 1/C)$

3.21.7 Narrow-face split loose flanges

Loose flanges of the type shown in Figure 3.21.3(k) may be of a split design to permit installation after heat treatment of the vessel or, in other cases, where it is desired to have the flanges completely removable from the vessel or nozzle.

Loose flanges split across a diameter and designed in accordance with Clause 3.21.6 may be used under the following provisions:

(a) Where the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange, i.e. without splits, using 200 percent of the total moment M_o as defined in Clause 3.21.6.5.

(b) Where the flange consists of two split rings, each ring shall be designed as if it were a solid flange, i.e. without splits, using 75 percent of the total moment M_o as defined in Clause 3.21.6.5. The pair of rings shall be assembled so that the splits in one ring shall be 90° from the splits in the other ring.

(c) The splits should preferably be midway between bolt holes.

3.21.8 Narrow-face non-circular shaped flanges with circular bore

These flanges shall be designed in accordance with Clauses 3.21.6 and 3.21.12, except that the outside diameter A for a non-circular flange with a circular bore shall be taken as the diameter of the largest circle, concentric with the bore, inscribed entirely within the outside edges of the flange. Bolt forces and moments, as well as stresses, are then calculated as for circular flanges, using a bolt circle drawn through the centres of the outermost bolt holes.

3.21.9 Flanges subject to external pressure

NOTE: When internal pressure occurs only during the required pressure test, the design may be based on external pressure and auxiliary devices such as clamps may be used during the application of the required test pressure.

3.21.9.1 *Design for external pressure*

The design of flanges for external pressure only shall be based on the equations given in Clause 3.21.6.6 for internal pressure except for the following:

For operating conditions—

$$M_o = H_D(h_D - h_G) + H_T(h_T - h_G) \qquad \text{. . . 3.21.9(1)}$$

A2 For gasket seating—

$$W = \frac{A_{m2} + A_b}{2} S_a \qquad \text{. . . 3.21.9(2)}$$

where

H_D = $0.785B^2P_e$

H_T = $H - H_D$

H = $0.785G^2\, P_e$

P_e = external design pressure, in megapascals

The flange shall also be designed to resist bolt loads due to $W = 0.5A_bR_{eb}$, without exceeding the criteria of Clause 3.21.6.7 where $f = 0.6R_e$ (flange material).

See Clause 3.21.6.7 for definitions of other symbols.

3.21.9.2 *Design for external and internal pressure*

When flanges are subject at different times during operation to external or internal pressure, the design shall satisfy the external pressure design requirements given in Clause 3.21.9.1 and the internal pressure design requirements given in Clause 3.21.6.

NOTE: The combined force of external pressure and bolt loading may plastically deform certain gaskets and result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized consideration should be given to gasket and facing details, so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

3.21.10 Flat-face flanges with metal-to-metal contact outside the bolt circle

The design of flat-face flanges with metal-to-metal contact outside the bolt circle shall comply with the relevant design method in ASME BPV VIII-1 or other methods agreed between the parties concerned.

3.21.11 Flanges with full-face gaskets

3.21.11.1 *General*

The flange design methods outlined in this Clause (3.21.11) are applicable to all circular flanges including loose type, integral type and hub type with full-face gaskets subject to internal pressure. Flanges for use with full-face gaskets are not recommended for use at pressures exceeding 2.1 MPa (see Clause 3.21.2(b)).

3.21.11.2 *Notation*

The notation for Clause 3.21.11 is the same as that of Clause 3.21.6.2, with the following modifications and additions:

b = effective gasket or joint-contact-seating width, in millimetres

$= \dfrac{C - B}{4}$

G = diameter at the location of that portion of the gasket reaction between the bolt circle and the internal diameter of the flange, in millimetres

$= C - 2h_G$

H = total hydrostatic end-force, in newtons

$= 0.785G^2P$

H_p = total joint-contact-surface compression force, in newtons

= addition of gasket-force between the bolt circle and the inside of the flange, plus the gasket-force between the bolt circle and the outside of the flange

$= (2b\pi GmP)\left(1 + \dfrac{h_G}{h'_G}\right)$

H_G = total gasket-force (difference between flange design bolt-force and total hydrostatic end-force), in newtons

$= W - H$

h_G = radial distance from the bolt circle to the reaction of that portion of the gasket-force between the bolt circle and the inside of the flange, in millimetres

$= \dfrac{(C - B)(2B + C)}{6(B + C)}$

h'_G = radial distance from the bolt circle to the reaction of that portion of the gasket force between the bolt circle and the outside of the flange, in millimetres

$= \dfrac{(A - C)(2A + C)}{6(C + A)}$

M_G = component of the internal moment at the bolt circle, due to gasket-force, in newton millimetres

$= \dfrac{(W - H)}{\dfrac{1}{h_G} + \dfrac{1}{h'_G}}$

W = flange design bolt-force for the operating conditions or gasket seating, as may apply (see Clause 3.21.11.4.3), in newtons

W_{m1} = minimum required bolt-force for operating conditions (see Clause 3.21.11.4.1(a)), in newtons

W_{m2} = minimum required bolt-force for gasket seating (see Clause 3.21.11.4.1(b)), in newtons

m = gasket factor, obtained from Table 3.21.11.4

y = gasket or joint-contact-surface seating stress, obtained from Table 3.21.11.4

Y' = factor involving K, obtained from Figure 3.21.6.6(A)

3.21.11.3 *Circular flange types*

For circular flange types, the classifications given in Clause 3.21.6.3 apply.

3.21.11.4 *Bolt forces*

3.21.11.4.1 *Required bolt-forces*

The flange bolt-force used in calculating the required cross-sectional area of bolts shall be determined in accordance with Items (a) and (b). In addition, the requirements of Clause 3.21.6.4.2 shall be met.

(a) *Force for operating conditions.* The required bolt-force for the operating conditions W_{m1} shall be sufficient to resist the hydrostatic end-force H, exerted by the maximum allowable working pressure on the area bounded by the diameter of that portion of the gasket reaction between the bolt circle and the inside of the flange, and in addition, to maintain on the gasket or joint-contact surface, a compression force H_p, which experience has shown to be sufficient to ensure a tight joint. (This compression force is expressed as a multiple, m, of the internal pressure. Its value is a function of the gasket material and facing. See Table 3.21.11.4.) The required bolt-force for the operating conditions, W_{m1}, is determined in accordance with the following equation:

$$W_{m1} = H + H_p = 0.785 G^2 P + 2b\pi GmP\left(1 + \frac{h_G}{h'_G}\right) + A_p m_p P \quad \ldots 3.21.11.4(1)$$

TABLE 3.21.11.4

GASKET MATERIALS AND SUGGESTED CONTACT FACTORS

Gasket material	Gasket factor, m	Min. design seating stress y, MPa
Soft rubber or neoprene	0.25	2.0
Rubber with fabric insertion	0.80	2.9
Compressed asbestos fibre	0.90	3.5

NOTE: Gasket factors for use with wide face flanges are not well known and are the subject of considerations for temperature, gasket characteristics, type of fluid, etc. Above are some suggested values for use with fluids below 260°C. Values which are too low may result in leakage at the joint without affecting the safety of the joint.

The effect of increasing these values will result in higher bolt stresses.

Asbestos type gaskets may not be permitted in some applications.

(b) *Gasket seating-force* Before a tight joint can be obtained, it is necessary to seat the gasket or joint-contact surface properly by applying a minimum initial force (under atmospheric temperature conditions without the presence of internal pressure), which is a function of the gasket material and the effective gasket area to be seated. The minimum initial bolt force required for this purpose (W_{m2}) shall be determined in accordance with the following equation:

$$W_{m2} = \frac{A_b Y_a}{2} > \pi bGy\left(1 + \frac{h_G}{h'_G}\right) \quad \ldots 3.21.11.4(2)$$

The need for providing sufficient bolt-force to seat the gasket or joint-contact-surface in accordance with Equation 3.21.11.4(2) will prevail in many low-pressure designs and with facings and materials that require a high seating-force, and where the bolt-force calculated by Equation 3.21.11.4(1) for the operating conditions is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pre-tighten the bolts to provide a bolt-force sufficient to satisfy both of these requirements, each one being individually investigated. When Equation 3.21.11.4(2) governs, flange proportions will be a function of the bolting instead of internal pressure.

3.21.11.4.2 *Total required and actual bolt areas* (A_m *and* A_b)

The requirements of Clause 3.21.6.4.3 apply.

3.21.11.4.3 *Flange design bolt-force* (W)

The requirements of Clause 3.21.6.4.4 apply.

3.21.11.5 *Flange moments*

In the calculation of flange stresses, the moment of force acting on the flange is the product of the force and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the force producing the moment. No credit shall be taken for reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof.

For the operating conditions, the total flange moment M_o, is the sum of only two individual moments M_D and M_T, as defined in Clause 3.21.6.2, and based on the flange design bolt-force of Equation 3.21.6.4.4(1) with moment arms as given in Table 3.21.11.5. In this calculation, it is assumed that when full fixation at the bolt circle is produced during bolting up, the gasket moments due to the reaction each side of the bolt circle are equal and opposite.

For the gasket seating conditions, the total flange moment, is based on the flange design bolt-force of Clause 3.21.11.4.3 which is opposed only by the gasket-force, in which case—

$$M_o = M_G = \frac{(W - H)}{\frac{1}{h_G} + \frac{1}{h'_G}} \quad \text{. . . 3.21.11.5}$$

TABLE 3.21.11.5

MOMENT ARMS FOR FLANGE FORCES UNDER OPERATING CONDITIONS

Values			
h_D	h_T	h_G	h'_G
$\frac{C-B}{2}$	$\frac{h_D + h_G}{2}$	$\frac{(C-B)(2B+C)}{6(B+C)}$	$\frac{(A-C)(2A+C)}{6(A+C)}$

3.21.11.6 *Calculation of flange stresses*

The stresses in the flange shall be determined for both the operating conditions and gasket seating, whichever controls, in accordance with the following equations:

Tangential flange stress—

$$S_T = \frac{Y' M_o}{t^2 B} \quad \text{. . . 3.21.11.6(1)}$$

Radial flange stress—

$$S_R = \frac{6M_o}{t^2(\pi C - nD)} \qquad \text{. . . 3.21.11.6(2)}$$

3.21.11.7 *Flange design strength*

The flange stress calculated by either equation in Clause 3.21.11.6 shall be no greater than S_f.

3.21.12 Reverse flange

3.21.12.1 *General*

The flange design methods outlined in this Clause (3.21.12) are applicable to reverse flanges of types shown in Figure 3.21.12.2.

The method is applicable where values of K is less than and equal to 2. For values of K greater than 2, the design method becomes increasingly conservative as values of K increase and the results shall be treated with caution.

3.21.12.2 *Notation*

The notation given in Clause 3.21.6.2, and illustrated in Figure 3.21.12.2, is used in the design methods in this Clause (3.21.12) with the following modifications and additions:

B = for reverse flanges, inside diameter of shell, in millimetres

B'' = inside diameter of reverse flange, in millimetres

d_r = factor which applies to reverse flanges, in millimetres to the third power

= $\frac{U_r}{V} h_{0r} g_0^2$

e_r = factor which applies to reverse flanges, in millimetres to the power of minus one

= $\frac{F}{h_{0r}}$

F = factor for integral type flanges (from Figure 3.21.6.6(B) substituting h_{0r} for h_0)

f = hub stress correction factor for integral flanges (from Figure 3.21.6.6(F)) substituting h_{0r} for h_0 for reverse flanges) (when greater than one, this is the ratio of the stress in the small end of the hub to the stress in the large end) (for values below the limit of the Figure, use $f = 1$)

h_{0r} = factor which applies to reverse flanges, in millimetres

= $\sqrt{(Ag_0)}$

K = ratio of outside diameter of flange to inside diameter of flange

= A/B'' for a reverse flange

L_r = factor which applies to reverse flanges

= $\frac{te_r + 1}{T_r} + \frac{t^3}{d_r}$

M_o = total moment acting upon the flange, for operating conditions or gasket seating as may apply, in newton millimetres (see Clause 3.21.6.5 and Clause 3.21.12.3); this notation applies to flanges covered by Clauses 3.21.6 to 3.21.9 inclusive, and Clause 3.21.12

S_{T1} = calculated tangential stress at outside diameter of a reverse flange, in megapascals

S_{T2} = calculated tangential stress at inside diameter of a reverse flange, in megapascals

T_r = factor which applies to reverse flanges

$$= \left(\frac{Z+0.3}{Z-0.3}\right)\alpha_r T$$

U_r = factor which applies to reverse flanges

= $\alpha_r U$

V = factor for integral type flanges (from Figure 3.21.6.6(C) substituting h_{0r} for h_0 for reverse flanges)

Y_r = factor which applies to reverse flanges

= $\alpha_r Y$ for flanges with ring type gaskets

= $\alpha_r Y'$ for flanges with full face gaskets

α_r = $\left(1+\frac{0.668(K+1)}{Y}\right)\frac{1}{K^2}$ for flanges with ring type gaskets

= $\left(1+\frac{0.668(K+1)}{Y'}\right)\frac{1}{K^2}$ for flanges with full face gaskets

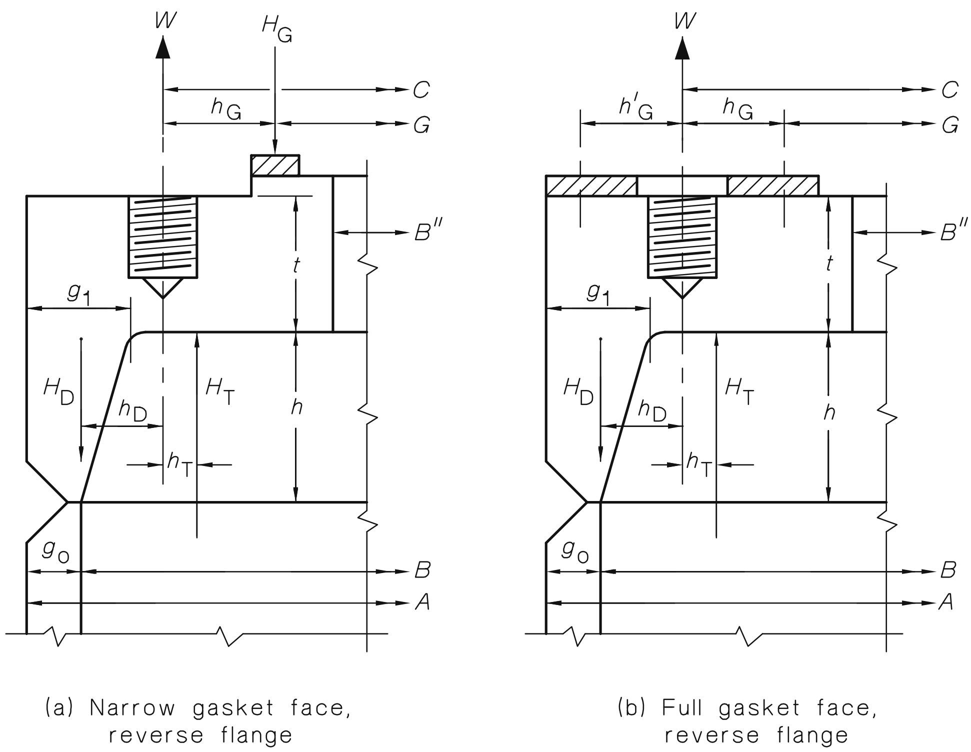


FIGURE 3.21.12.2 REVERSE FLANGE NOTATION

3.21.12.3 *Flange moments for reverse flanges with ring-type gaskets*

The total flange moment (M_o), shall be calculated for both the gasket seating and operating conditions in accordance with Clause 3.21.6.5, substituting Figure 3.21.12.2(a) where reference is made to Figure 3.21.6.2, and the following:

(a) For gasket seating condition—

$$M_o = Wh_G \quad \text{. . . 3.21.12.3(1)}$$

(b) For operating conditions—

$$M_o = M_D + M_T + M_G \quad \text{. . . 3.21.12.3(2)}$$

NOTE: For reverse flanges, h_D and H_T are negative (see Figure 3.21.12.2(a)).

If (M_o) is negative, its absolute value shall be used in calculating stresses for comparison with allowable stresses.

3.21.12.4 *Flange moments for reverse flanges with full-face gaskets*

The total flange moment (M_o) shall be calculated for both the gasket seating and operating conditions in accordance with Clause 3.21.11.5 and Figure 3.21.12.2(b), and the following:

(a) For gasket seating conditions—

$$M_o = M_G \quad \text{. . . 3.21.12.4(1)}$$

(b) For operating conditions—

$$M_o = M_D + M_T \quad \text{. . . 3.21.12.4(2)}$$

NOTE: For reverse flanges, h_D and H_T are negative (see Figure 3.21.12.2(b)).

h_T maybe positive as in Figure 3.21.12.2(a) but can be negative if the line of action of H_T is on the other side of the bolt circle.

If M_o is negative, its absolute value shall be used in calculating stresses for comparison with allowable stresses.

3.21.12.5 *Calculation of flange stresses*

The stresses in the flange shall be determined for both the gasket seating and operating conditions in accordance with the following:

(a) Stresses at flange outside diameter—

$$S_H = \frac{f\ M_o}{L_r\ g_1^2\ B''} \quad \text{. . . 3.21.12.5(1)}$$

$$S_R = \frac{(1.33\,te_r + 1)\,M_o}{L_r\ t^2\ B''} \quad \text{. . . 3.21.12.5(2)}$$

$$S_{T1} = \frac{Y_r\ M_o}{t^2\ B''} - ZS_R\left(\frac{0.67\ te_r + 1}{1.33\ te_r + 1}\right) \quad \text{. . . 3.21.12.5(3)}$$

(b) Stresses at flange inside diameter—

$$S_{T2} = \left(\frac{M_o}{t^2\ B''}\right)\left[Y - \frac{2K^2\left(1 + \frac{2\,te_r}{3}\right)}{(K^2 - 1)\,L_r}\right] \text{ for flanges with ring type gaskets} \quad \text{. . . 3.21.12.5(4)}$$

$$S_{T2} = \left(\frac{M_o}{t^2\ B''}\right)\left[Y' - \frac{2K^2\left(1+\frac{2te_r}{3}\right)}{\left(K^2-1\right)L_r}\right] \text{ for flanges with full face gaskets} \quad \ldots 3.21.12.5(5)$$

NOTE: For simplicity, the designer may calculate the construction as a loose-type flange provided that none of the following values are exceeded:

P = 2.1 MPa, Design temperature = 370°C.

In this case, the stress at the flange outside diameter is $S_{T1} = Y_r\ M_o/(t^2B'')$ and at the inside diameter is $S_{T2} = Y'M_o/(t^2B'')$. S_H and $S_R = 0$.

3.21.12.6 *Flange design strength*

The flange stresses calculated by the equations in Clause 3.21.12.5 shall not exceed the permissible stresses specified in Clause 3.21.6.7.

3.22 PIPES AND TUBES

3.22.1 General

The design of pipe and tube components shall be accordance with AS 4041 modified by Clauses 3.22.2 and 3.22.3 for those which are integral components of a vessel. Where the rules of AS 4041 are used for non-standard piping component design for pressure vessels, the design stresses of this Standard (AS 1210) shall be used rather than those in AS 4041.

3.22.2 Thickness

The calculated wall thickness for tube and pipe shall be determined in accordance with—

(a) Clause 3.7, when subject to internal pressure; and

(b) Clause 3.9, when subject to external pressure.

Additional thickness shall be provided in accordance with Clause 3.4.2. Where the tube end is threaded, the tube thickness shall be based on the bottom of the thread.

Where the tube is bent, the resulting thickness at the thinnest part shall be no less than that required for straight tube unless it can be demonstrated that the method of forming the bend results in no decrease in strength at the bend compared with straight tube.

For staytubes, see Clause 3.16.5, and for tubes in heat exchangers, see Clause 3.17.

3.22.3 Attachment

Attachment of tubes and pipes to shell and ends shall be in accordance with Clause 3.19, and staytubes shall be attached to stayed surfaces in accordance with Clause 3.16.5.

Attachment of tubes to flat tubeplates or other surfaces shall be in accordance with Clause 3.17.

3.23 JACKETED VESSELS

3.23.1 General

Jacketed vessels, including jacketed troughs, shall be designed in accordance with the requirements given for each element given elsewhere in this Standard except where modified in this Clause (3.23). The jacketed portion of the vessel is defined as the inner and outer walls, the closure devices, and all other penetrations or parts within the jacket that are subjected to pressure stresses. Parts such as nozzles, closure members and stiffening or stay rings are included.

The inner vessel shall be designed to resist the full differential pressure that may exist under any operating condition, including accidental vacuum in the inner vessel due to condensation of vapour contents where this circumstance can arise.

Where the inner vessel is to operate under vacuum and the hydrostatic test pressure for the jacket is correspondingly increased to test the inner vessel externally, care shall be taken that the jacket shell is designed to withstand this extra pressure.

The effect of localized internal and external forces and thermal expansion shall be considered. If the number of full thermal stress cycles is expected to exceed 5000, the design shall cater for thermal stresses caused by varying expansion rates between the jacket and inner vessel.

Impingement plates or baffles shall be provided at the jacket inlet where erosion of the vessel or jacket wall is a possibility due to condensation of steam or other condensable vapour.

In areas of high local stress concentration a detailed stress analysis shall be required except that in simple cases and where the temperature differential is low or where there is good evidence of satisfactory past experience, Clause 3.23 and design strengths given in Table B1(I) may be used for Class 1H and 2H construction.

3.23.2 Types of jacketed vessels

This Clause (3.23) applies to jacketed vessels having jackets that cover the shell or ends as illustrated in Figure 3.23.2 and partial jackets as illustrated in Figure 3.23.7. Jackets, as shown in Figure 3.23.2, shall be continuous circumferentially for Types 1, 2, 4 and 5 shown, and shall be circular in cross-section for Type 3. The use of a combination of the types shown is permitted on a single vessel provided that the individual requirements for each are met. Dimpled jackets are not covered by this Clause (see Clause 3.16.6). (For jacketed troughs see Clause 3.23.8.)

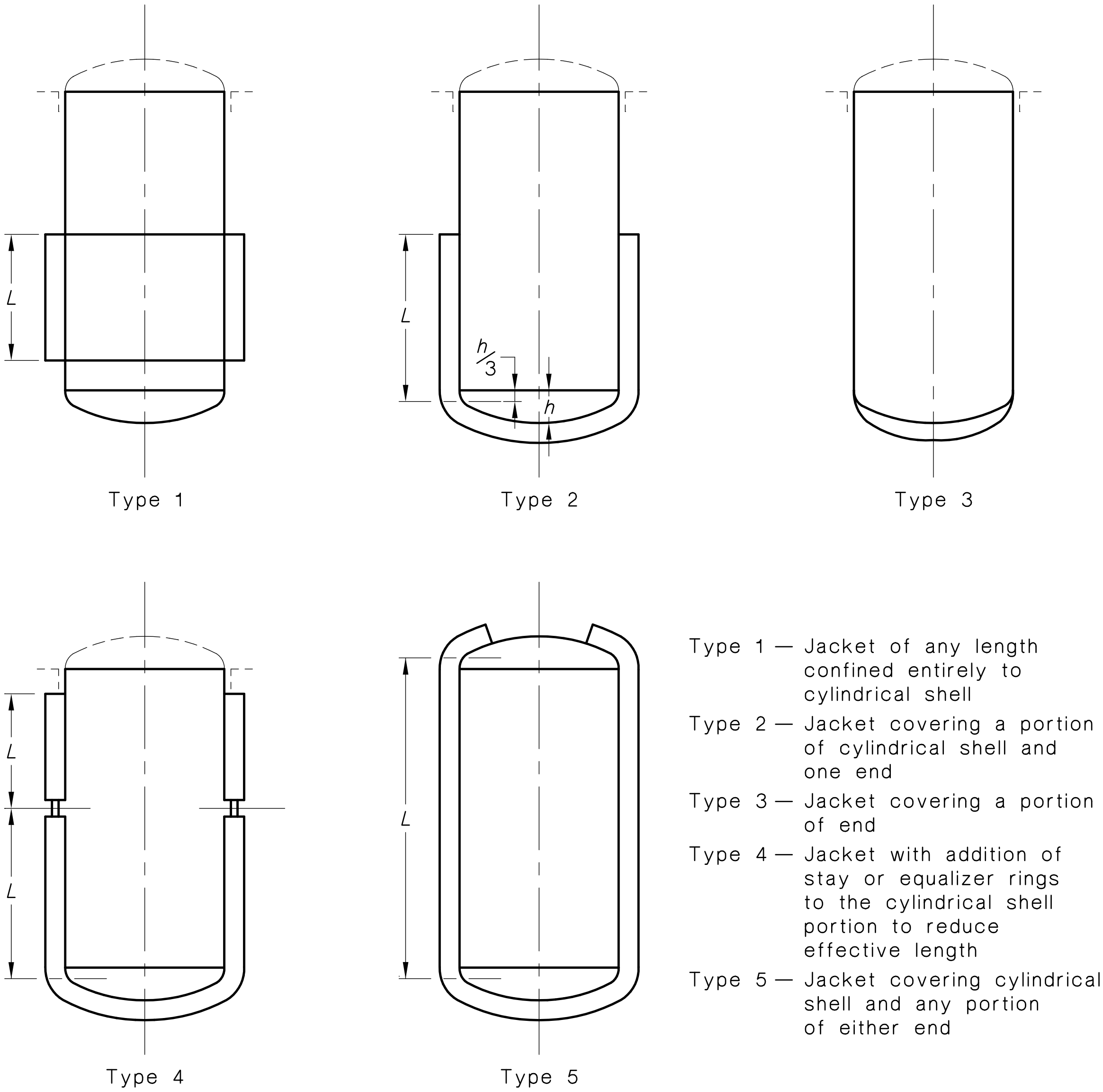


FIGURE 3.23.2 SOME ACCEPTABLE TYPES OF JACKETED VESSEL

3.23.3 Design of jacket shells and jacket ends

The design of jacket shells and jacket ends shall comply with the requirements of Section 3 of this Standard, and with the general requirements of Clause 3.23.1.

3.23.4 Notation

For the purpose of this Clause (3.23), the following notation applies:

t_s = actual thickness of inner vessel wall, in millimetres

t_{rj} = minimum required thickness of outer jacket wall exclusive of corrosion allowance, in millimetres

t_{rc} = minimum required thickness exclusive of corrosion allowance of closure member as determined herein, in millimetres

t_c = actual thickness of closure member, in millimetres

t_j = actual thickness of outer jacket wall, in millimetres

t_n = nominal thickness of nozzle, in millimetres

r = corner radius of torus closures, in millimetres

R_s = outside radius of inner vessel, in millimetres

R_j = inside radius of jacket, in millimetres

R_p = radius of opening in the jacket at the jacket penetration, in millimetres

P = design pressure in the jacket chamber, in megapascals

P_v = design vacuum in inner vessel, in megapascals

f = design strength, in megapascals

j = jacket space, in millimetres

= inside radius of jacket, minus outside radius of inner vessel, in millimetres

a, b, c, Y, Z = minimum weld dimensions for attachment of closure member to inner vessel measured as shown in shown in Figures 3.23.5 and 3.23.6, in millimetres

L = design length of a jacket section as shown in Figure 3.23.2, in millimetres.

This length is determined as—

(a) the distance between inner vessel end bend lines plus one-third of the depth of each inner vessel end where there are no stiffening rings or jacket closure between the end bend lines;

(b) the centre-to-centre distance between two adjacent stiffening rings or jacket closures; or

(c) the distance from the centre of the first stiffening ring or the jacket closure to the jacketed inner end bend line plus one-third of the inner vessel end, all measured parallel to the axis of the vessel.

For the design of a closure member or stiffening ring, the greater adjacent L shall be used.

3.23.5 Design of jacket closures

Jacket closures shall conform to those shown in Figure 3.23.5, and shall comply with the following requirements unless otherwise agreed between the parties concerned.

(a) Closures of the type shown in Figure 3.23.5(a) that are used on Type 1, Type 2, or Type 4 jacketed vessels as shown in Figure 3.23.2 shall have t_{rc} at least equal to t_{rj} and corner radius r shall not be less than $3t_c$. This closure design is limited to a maximum thickness t_{rc}, of 15 mm. Where this construction is used on Type 1 jacketed vessels, the weld dimension Y shall be not less than $0.7t_c$; and where used on Types 2 and 4 jacketed vessels, the dimension Y shall be not less than $0.85t_c$.

(b) Closures of the type shown in Figure 3.23.5 (b-1) and (b-2) shall have t_{rc} at least equal to t_{rj}. A butt weld attaching the closure to the inner vessel and fully penetrating the closure thickness t_c, may be used with any of the types of jacketed vessels shown in Figure 3.23.2. However, a fillet weld having a minimum throat dimension of $0.7t_c$ may also be used to join the closure of the inner vessel of Type 1 jacketed vessels of Figure 3.23.2.

(c) Closures of the type shown in Figure 3.23.5(c) shall be used only on Type 1 jacketed vessels shown in Figure 3.23.2. The closure thickness t_{rc}, shall be determined by Clause 3.10 but shall be not less than t_{rj}. The angle α shall be limited to 30° maximum.

(d) Closure of the types shown in Figure 3.23.5(d-1), (d-2), (e-1), and (e-2), shall be used only on Type 1 jacketed vessels as shown in Figure 3.23.2 and with the further limitation that t_{rj} does not exceed 15 mm. The required minimum thickness for the closure bar shall be the greater value of that determined by the following Equation:

$$t_{rc} = 2(t_{rj}) \qquad \ldots 3.23.5(1)$$

$$t_{rc} = 0.707j\left(\frac{P}{f}\right)^{0.5} \qquad \ldots 3.23.5(2)$$

Fillet weld sizes shall be as follows:

(i) Y shall be not less than the smaller of $0.75t_c$ and $0.75t_s$.

(ii) Z shall be not less than t_j.

(e) Closure bar and closure bar to inner vessel welds of the types shown in Figure 3.23.5(f-1), (f-2) and (f-3) may be used on any of the types of jacketed vessels shown in Figure 3.23.2. For all other types of jacketed vessels the required minimum closure bar thickness shall be determined by the following equation:

$$t_{rc} = 1.414\left(\frac{PR_s j}{f}\right)^{0.5} \qquad \ldots 3.23.5(3)$$

The width of the jacket space shall not exceed the value determined by the following equation:

$$j = \frac{2ft_s^2}{PR_j} - 0.5(t_s + t_j) \qquad \ldots 3.23.5(4)$$

Weld sizes connecting the closure bar to the inner vessel shall be as follows:

(i) Y shall be not less than the smaller of $1.5t_c$ and $1.5t_s$ and shall be measured as the sum of dimensions a and b as shown in the appropriate sketch of Figure 3.23.5.

(ii) Z minimum fillet leg length necessary when used in conjunction with a groove weld or another fillet weld to maintain the minimum required Y dimension.

(f) Jacket to closure bar attachment welds shown in Figure 3.23.5(g-1), (g-2) and (g-3) may be used on any of the types of jacketed vessels shown in Figure 3.23.2. Attachment welds shown in Figure 3.23.5(g-4) may be used on any of the types of jacketed vessels shown in Figure 3.23.2 where t_{rj} does not exceed 15 mm. Attachment welds shown in Figure 3.23.5(g-5) and (g-6), may be used in Type 1 jacketed vessels shown in Figure 3.23.2 where t_j does not exceed 15 mm.

(g) Closures shown in Figure 3.23.5(h) and (j) shall be limited to jackets where t_{rj} does not exceed 15 mm.

(h) Closures for conical or toriconical jackets shown in Figure 3.23.5(k) and (l) shall comply with the requirements of Type 2 jacketed vessels shown in Figure 3.23.2.

(i) Each radial weld in a closure member shall be a butt-welded joint penetrating through the full thickness of the member and shall be ground flush where attachment welds are to be made.

(j) Closures for any type of staybolted jacket may be designed in accordance with the requirements of Type 1 jackets shown in Figure 3.23.2 provided that the entire jacket is staybolted to compensate for pressure end-forces.

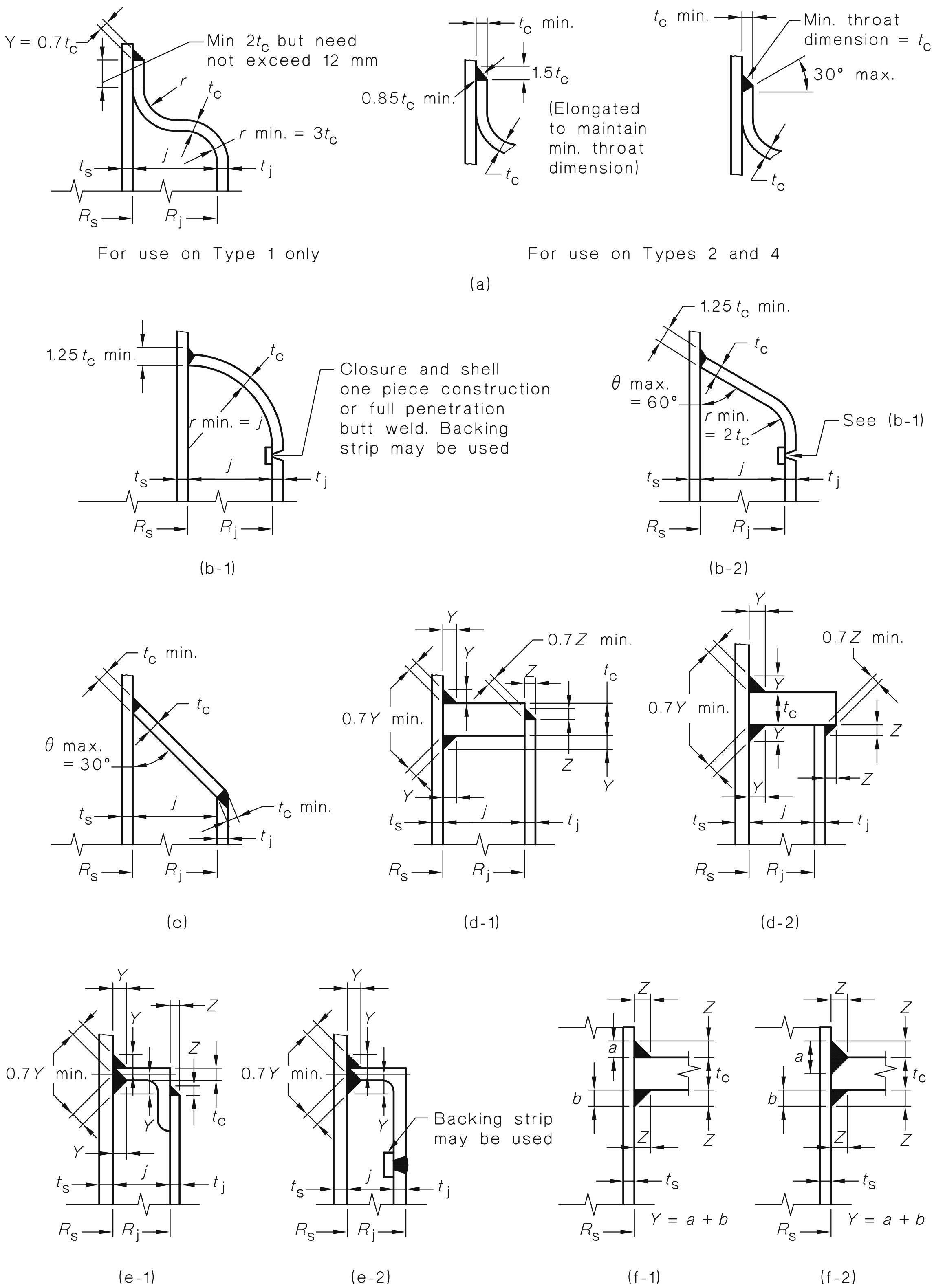


FIGURE 3.23.5 (in part) SOME ACCEPTABLE TYPES OF JACKET CLOSURE

(See Clause 3.23.5 for limitations on use)

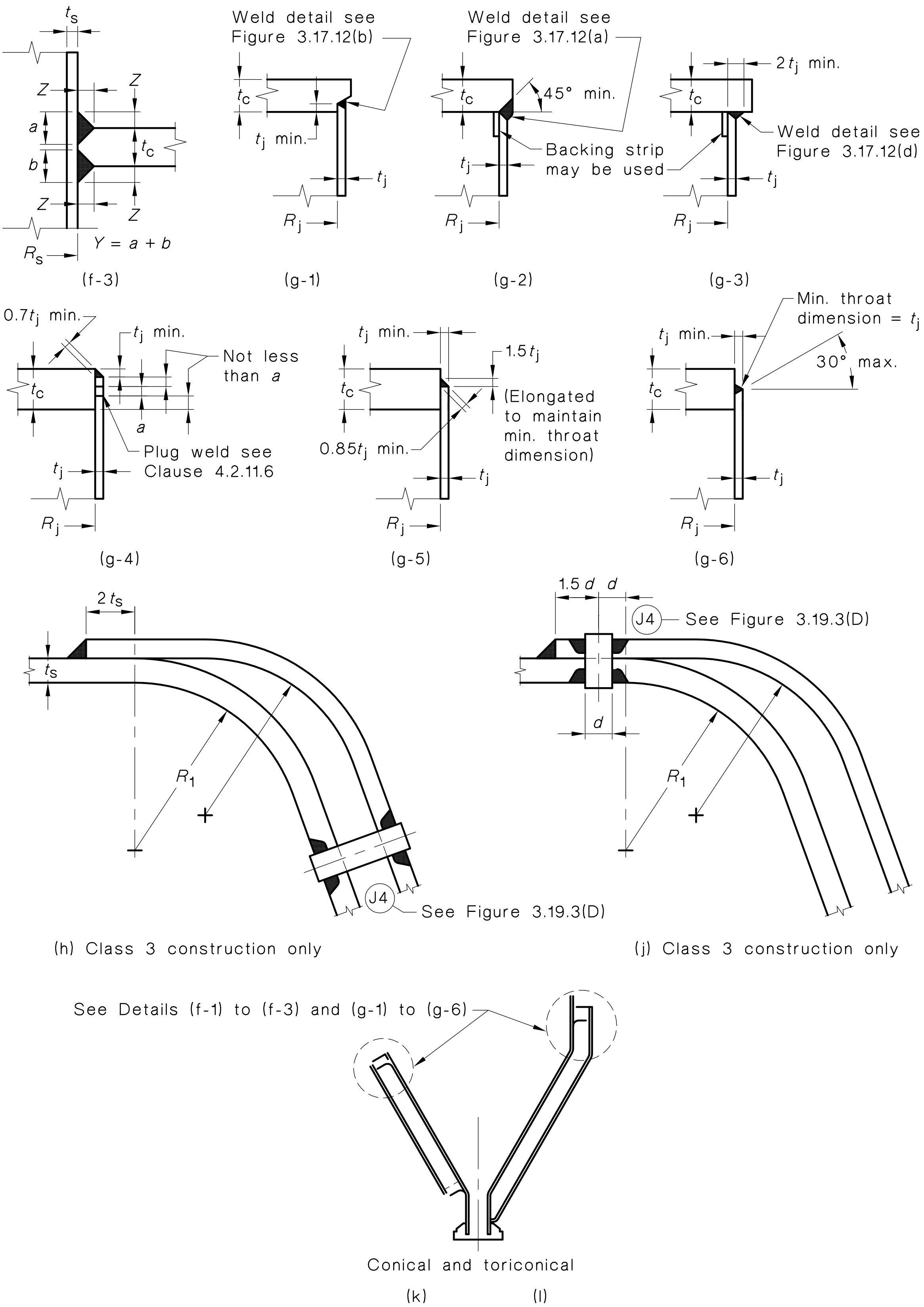


FIGURE 3.23.5 (in part) SOME ACCEPTABLE TYPES OF JACKET CLOSURES

(See Clause 3.23.5 for limitations on use)

3.23.6 Design of penetrations through jackets

The following requirements apply to openings through jackets:

(a) The design of openings through the jacket space shall be in accordance with the requirements of this Standard.

(b) Reinforcement of the opening in the jacket shall not be required for penetrations shown in Figure 3.23.6 since the opening is stayed by virtue of the nozzle or neck of the closure member.

(c) The jacket penetration closure member minimum thickness considers only pressure membrane loading. Axial pressure loadings and secondary loadings given in Clause 3.2.3 shall be considered in the design (see Clause 3.23.6(d)(vi)).

(d) Jacket penetration closure member designs shown in Figure 3.23.6 shall conform to the following requirements:

(i) Where the nozzle is used as the closure member as shown in Figure 3.23.6(a), the jacket shall be welded to the nozzle.

(ii) The minimum required thickness t_{rc}, for designs Figure 3.23.6(b) and (d) shall be calculated as a shell under external pressure in accordance with Clause 3.9.

(iii) The minimum required thickness t_{rc}, for design Figure 3.23.6(c) shall be equal to t_{rj}.

(iv) For designs Figure 3.23.6(e-1) and (e-2), the thickness required of the closure member attached to the inner vessel t_{rc1}, shall be calculated as a shell under external pressure in accordance with Clause 3.9. The required thickness of the flexible member t_{rc2}, shall be determined by one of the following equations:

When no tubular section exists between jacket and torus—

$$t_{rc2} = \frac{Pr}{(f\eta - 0.5P)} \qquad \text{. . . 3.23.6(1)}$$

When tubular sections exists between jacket and torus—

$$t_{rc2} = \frac{PR_p}{(f\eta - 0.5p)} \qquad \text{. . . 3.23.6(2)}$$

where

η = welded joint efficiency from Table 3.5.1.7 for circumferential weld in the torus for equation using r, or for any weld in opening closure member for equation using R_p, radius of penetration.

(v) The minimum thickness t_{rc}, for design Figure 3.23.6(f) shall be calculated as a shell of radius R_p, under external pressure in accordance with Clause 3.9.

(vi) Designs in Figure 3.23.6(b), (c), (d) and (e) provide for some flexibility and are designed on a similar basis to that of expansion joints under the conditions of Clause 3.1.3 in combination with Clauses 3.2.3 and 3.3.1. Only pressure membrane loading is considered in establishing the minimum thickness of the penetration closure member, and it is not the intent that the combination of direct localization and secondary bending stress need be held to the design strength values in Clause 3.3.1. It is recognized that high localized and secondary bending stresses may exist.

(vii) All radial welds in opening sealer membranes shall be butt-welded joints penetrating through the full thickness of the member.

(viii) Closure member wells shall be circular, elliptical, or obround in shape where possible. Rectangular member wells are permissible provided that corners are rounded to a suitable radius.

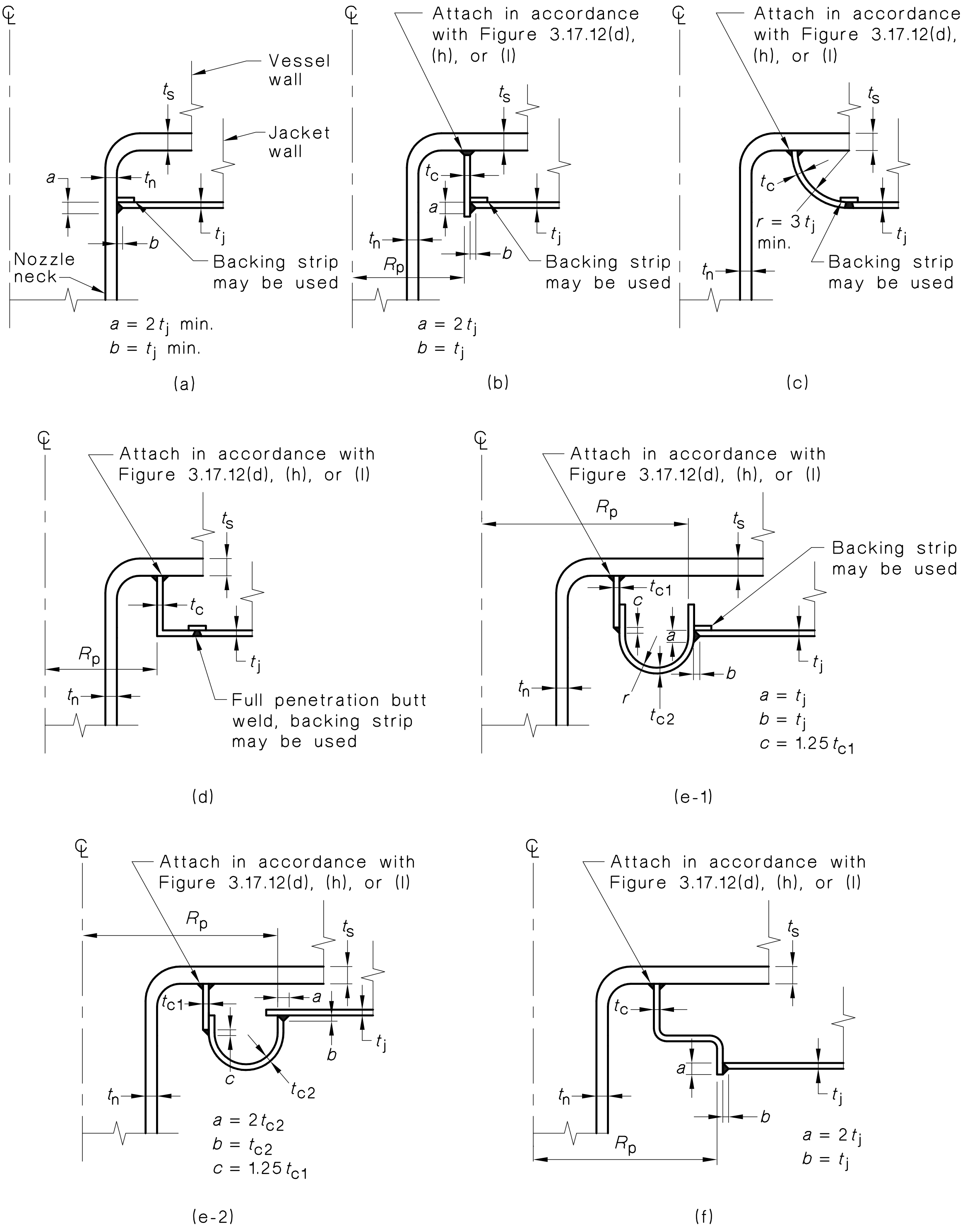


FIGURE 3.23.6 SOME ACCEPTABLE TYPES OF PENETRATION DETAILS

3.23.7 Design of partial jackets (excluding troughs)

3.23.7.1 *General*

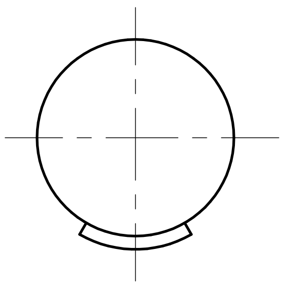
Partial jackets are jackets that encompass less than the full circumference of the vessel. Some variations are shown in Figure 3.23.7.

3.23.7.2 *Application*

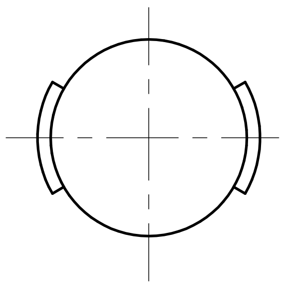
The requirements for construction of jacketed vessels given in Clauses 3.23.1 to 3.23.6 shall apply to partial jackets with the following exceptions:

(a) Stayed partial jackets shall be designed and constructed in accordance with Clause 3.16. Closure members shall comply with Clause 3.23.5.

(b) Partial jackets that, by virtue of their service or configuration, do not lend themselves to staybolt construction may be fabricated by other means provided that they are designed using appropriate stress values and are proof hydrostatic tested in accordance with Clause 5.12.



a) Continuous partial jacket



b) Multiple or pod type jacket

FIGURE 3.23.7 SOME TYPES OF PARTIAL JACKET

3.23.8 Jacketed troughs

The design of open-top trough-shaped vessels with jacket spaces for steam heating, unless otherwise agreed by the parties concerned shall comply with one of the following:

(a) *As shown in Figure 3.23.8(a)* The thickness of the flat portion of the side-plates subject to stream pressure shall be calculated as for a flat surface, and the side-plates shall be provided with stay-bolts in accordance with Clause 3.16. The inner bottom half cylindrical plate shall be stiffened or stayed as for a cylinder in compression, the outer half cylindrical plate being calculated as for a cylinder in tension. The top edges of the jacket side-plates shall be butt-welded to a longitudinal bar taking the pressure load. The troughs shall be suitably stiffened to resist distortion due to out of balance loads, e.g. by tie bars.

(b) *As shown in Figure 3.23.8(b)* The design shall comply with the requirements of Clause 3.23.8(a), except that the jacket side-plates shall be flanged over at the top and welded to the jacket and inner plates to take the pressure load.

(c) *As shown in Figure 3.23.8(c)* The whole area of the plate subject to pressure shall be stayed as flat surfaces in accordance with Clause 3.16. The troughs shall be suitably stiffened to resist distortion due to out of balance loads, e.g. by tie bars.

(d) *As shown in Figure 3.23.8(d)* Shallow curved trough plates shall be calculated as flat surfaces throughout, and shall be provided with stay-bolts in accordance with Clause 3.16.

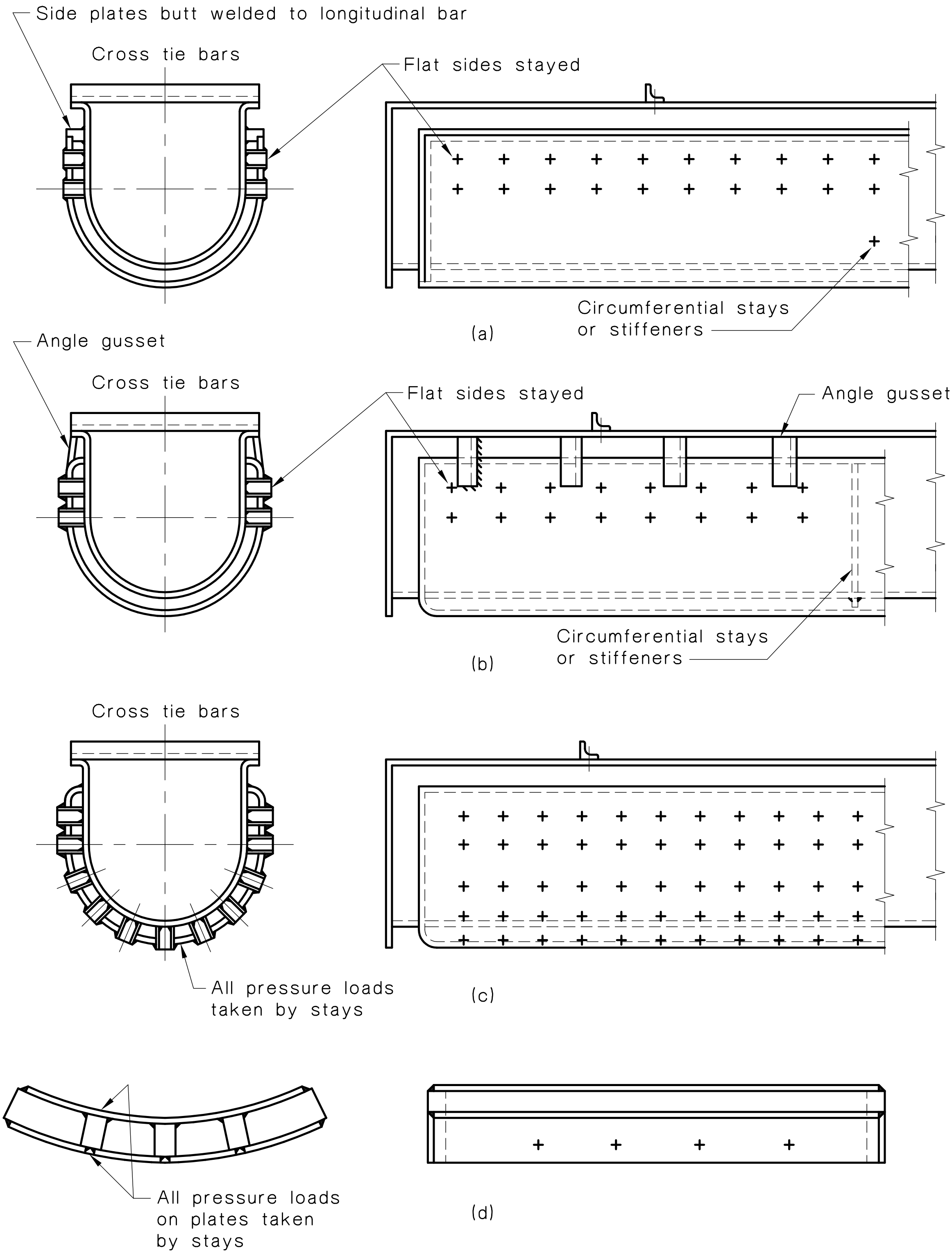


FIGURE 3.23.8 STEAM PRESSURE JACKETED TROUGHS

3.24 VESSEL SUPPORTS

3.24.1 General

Vessels shall be so supported and the supporting members arranged, or attached to the vessel wall, or both, in such a way as will withstand the maximum imposed loadings (see Clause 3.2.3), without causing excessive localized stresses and deformations in the vessel wall and instability of the vessel.

Supports shall be designed to allow for movement of the vessel wall due to thermal and pressure changes, and also for the possibility that highest stress may occur in some vessels under hydrostatic tests before pressure is applied. Care shall be taken that the temperature gradients in external structures immediately adjacent to the shell do not produce stresses in excess of those laid down as permissible. If necessary, lagging should be applied to limit the temperature gradient to a value producing acceptable stresses.

Full details for the design of supports and attachments are not given because they involve many variables, such as the size, weight, service temperature and pressure, the arrangement of the supporting structure, and any loads due to piping, or the like, attached to the vessel. Where the proposed supports depart from normal or proven practice or reasonable doubts exist as to their suitability, the design shall be assessed by detailed analysis as set out in Appendix N.

Typical vessel supports are shown in Figure 3.24. See Clause 3.26.10 for supports for transportable vessels.

3.24.2 Supporting members

The design of supporting members (including brackets, columns, etc.) and anchors shall conform to good structural practice.

Steel supports that do not form part of the vessel shall comply with AS 3990 or AS 4100. Reinforced concrete supports shall comply with AS 3600.

Suitable means shall be provided to prevent corrosion between the vessel wall and the supporting members. Fire-resistant supports shall be used in environments where fire hazards may occur and also for a vessel with flammable content.

Substantial foundations shall be provided so as to withstand the maximum imposed loading (see Clause 3.2.3) and to prevent subsidence or settling which may result in excessive loading of the vessel.

3.24.3 Supports for vertical vessels

3.24.3.1 *Bracket support*

(See Figure 3.24(a).) Where vertical vessels are supported by brackets or lugs attached to the shell, the supports under the bearing surfaces shall be as close to the shell as clearance for any insulation will permit. The choice between a number of brackets and a ring girder will depend upon the conditions for each individual vessel.

3.24.3.2 *Column support*

(See Figure 3.24(b).) Vertical vessels supported on a number of posts or columns may require bracing or stiffening by means of a ring girder, internal partition or similar device, to resist the forces tending to buckle the vessel wall.

3.24.3.3 *Skirt support*

(See Figure 3.24(c) and (d).) Vertical vessels may be supported on cylindrical or conical skirts which attach to the cylindrical portion of the vessel and this method is recommended for large vessels.

Where the product of a skirt diameter (millimetres), thickness (millimetres), and temperature at the top of the skirt above ambient (°C) exceeds 16×10^6, account shall be taken of the discontinuity stresses in both skirt and vessel induced by the temperature gradient in the upper section of the skirt. Skirts shall be designed to avoid buckling, and otherwise comply with the requirements of Clause 3.7.5.

Skirts shall have a half apex angle (α) not greater than 30°.

Openings in the skirt (see Clause 3.24.8) shall be reinforced if necessary.

3.24.3.4 *Stool support*

(See Figure 3.24(e).) Vertical vessels may be supported on stools. Stools shall be designed in accordance to Clause 3.24.3.3 with particular attention given to the stresses in the end at the attachment where a decrease in the stool diameter will increase the unit reaction load normal to the end surface. The design of the vessel and stool shall consider the worst combination of design and superimposed loads expected in service (see Clause 3.2.3) and during the initial hydrostatic testing, consideration should also be given to future hydrostatic testing on location.

3.24.3.5 *Skirt or stool inspection opening*

Such openings shall have suitable reinforcement or self-reinforcement to satisfy the requirements of Clause 3.18.

3.24.4 Supports for horizontal vessels

(See Figures 3.24(f) and (g).) Horizontal vessels may be supported by means of saddles, equivalent leg support, ring supports or suspension members. Vessels exceeding 1 m diameter shall be provided with saddle supports subtending at least 120° shell circumference continuously or shall be supported by other means proven to be suitable by analysis (see Clause 3.24.1).

Supports should be as few as possible, preferably two in the length of the vessel. Where this is not practicable, provision shall be made to ensure suitable distribution of load. The vessel may be stiffened where necessary by stiffening rings at intermediate sections.

With thin-walled vessels, vacuum vessels, or with large horizontal storage vessels that may distort excessively due to the vessel weight when internal pressure nears atmospheric, consideration shall be given to the placing of supports near the ends of the vessel or using ring supports, stiffeners or other reinforcements to prevent stresses in the shell in excess of those permitted and to avoid excessive distortion.

Ring supports shall be calculated by the following equation:

$$f = \frac{K_1 W' R}{Z} + \frac{K_2 W'}{A_s} \qquad \text{. . . 3.24.4}$$

where

A_s = cross-sectional area of the section

f = design strength at calculation temperature, in megapascals (Table B1)

W' = load acting on one ring, in newtons

R = radius of the ring measured to the neutral axis, in millimetres

Z = section modulus of the cross-section of the ring support, in millimetres to the third power; in the calculation of Z and R a part of the shell may be included having an effective length, L_s, as defined in Clause 3.9

K_1, K_2 = factors dependent on half of the included angle of the supports θ (see Figure 3.24(g) and Table 3.24.4).

Welds attaching ring supports to the vessel shall have a minimum leg length no less than the thickness of the thinner of the shell and the web of the ring.

Where local stresses at the supports are relevant to the design, Appendix N shall be used.

TABLE 3.24.4

ANGLE FACTORS FOR RING SUPPORT

Angle θ°	K_1	K_2
30	0.075	0.41
35	0.065	0.40
40	0.057	0.39
45	0.049	0.38
50	0.043	0.37
55	0.039	0.36
60	0.035	0.35
65	0.030	0.34
70	0.025	0.32
75	0.020	0.31
80	0.017	0.29
85	0.015	0.27
90	0.015	0.25

3.24.5 Supports for vessels subject to external pressure

Vessels subject to external pressure shall be supported through the medium of a substantially continuous ring or equivalent means to limit distortion.

NOTE: Concentrated loading on shell or ends of such vessels may cause deformations which seriously reduce the buckling resistance of the vessel.

3.24.6 Supports for jacketed vessels

Where vessel supports are attached to the jacket, consideration shall be given to the transfer of the supported load of inner vessel and contents.

3.24.7 Attachment of supports

Where supports are attached to vessels, the attachment shall be in accordance with Clause 3.25 and shall be located to leave all circumferential welds clear for inspection unless otherwise approved.

If a doubler plate is used and it either covers a vessel butt weld or the vessel will be subject to post weld heat treatment, the doubler plate should be provided with at least one tell-tale hole.

NOTE: There might be other reasons to include a tell-tale hole, such as to act as a breather during welding or to aid in the detection of possible cracking if the vessel is subject to cyclic thermal operation.

Attachments to pressure vessels with directly mounted reciprocating machines shall be designed and installed in such a manner as to avoid fatigue cracking at such attachments. Where vibration from any source may induce fatigue failure in the pressure parts of a vessel at the attachment of the vessel supports or equipment (e.g. air compressors) mounted on the vessel, provision shall be made to adequately distribute the load of the attachment. Where doubling plates are used they shall comply with Clause 3.26.10.2.

3.24.8 Access for inspection

Supports shall be designed to facilitate inspection of the vessel. Openings shall be made in the sides of skirts or stools if the bottoms are not readily visible through the supporting structure. Saddle supports not seal welded shall be designed to permit inspection of the shell wall on the saddle.

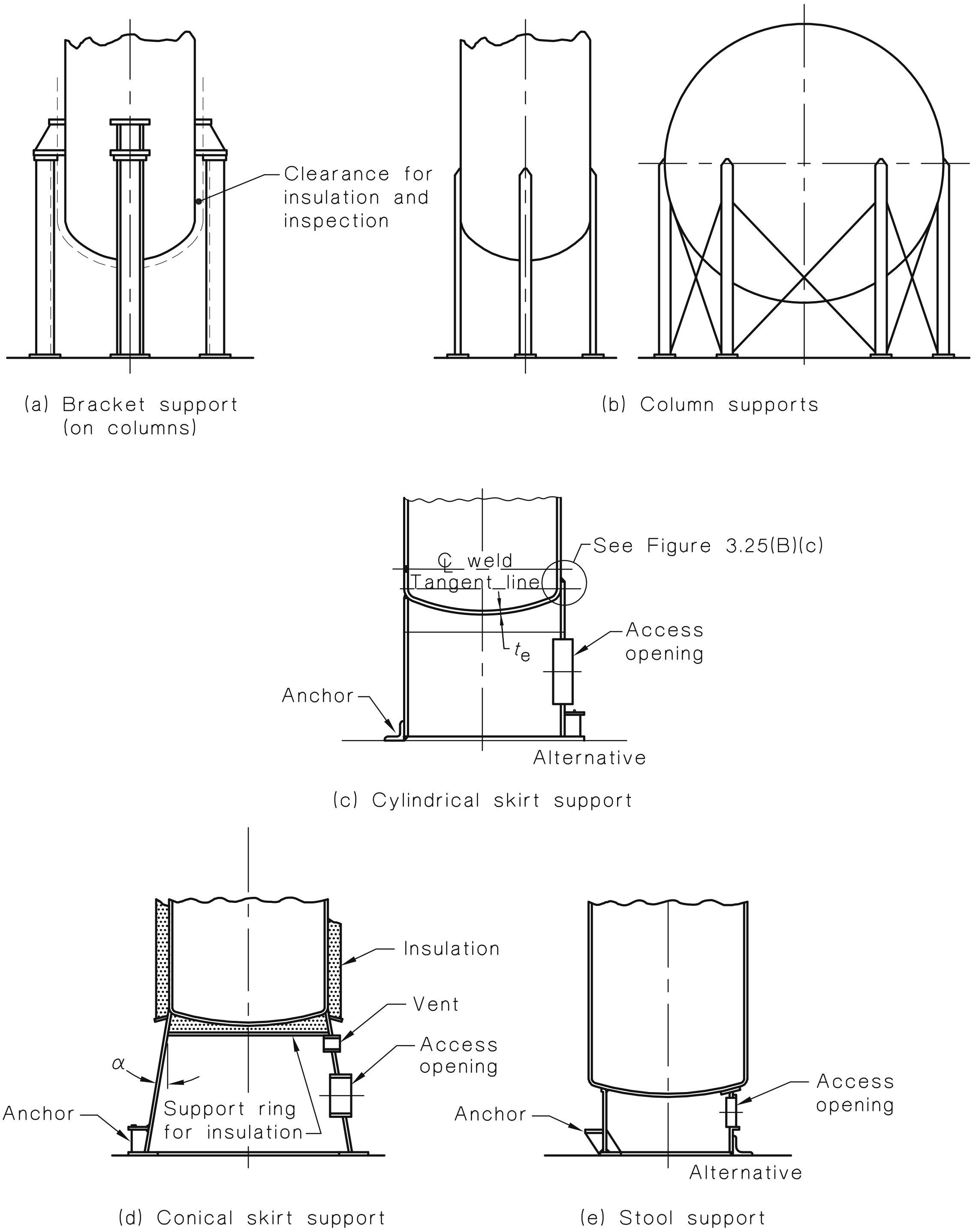


FIGURE 3.24 (in part) SOME TYPICAL VESSEL SUPPORTS

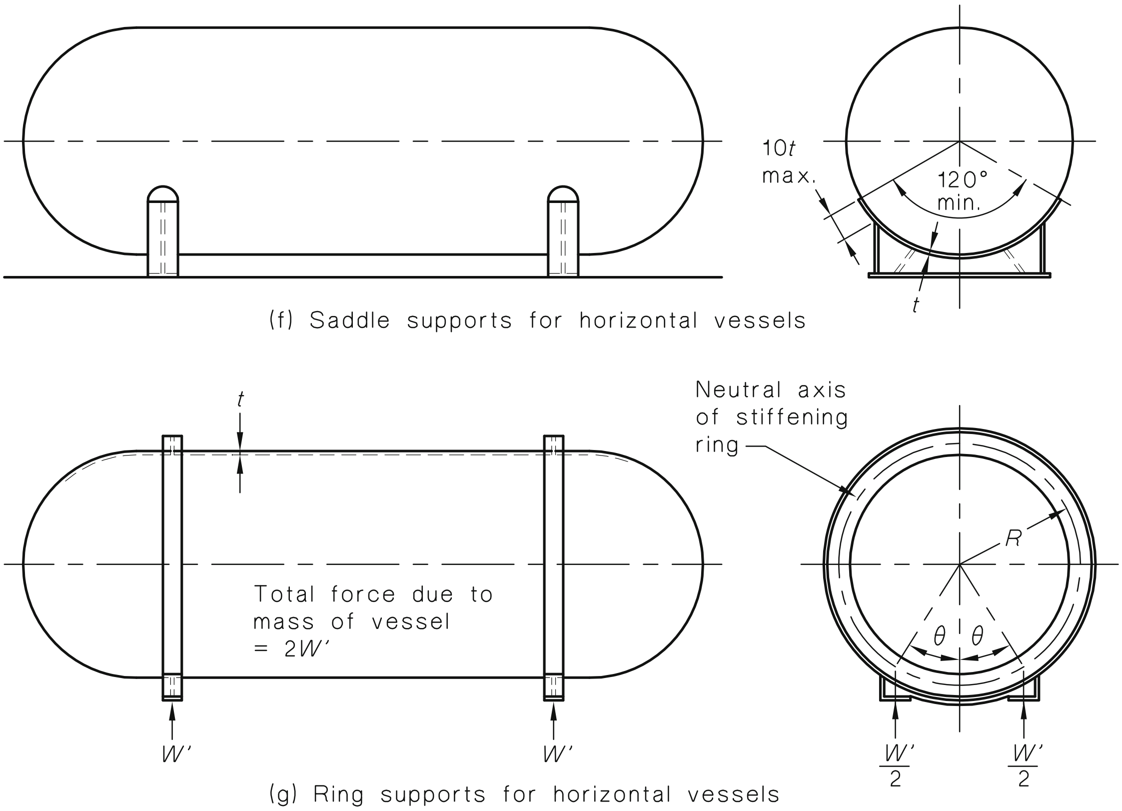


FIGURE 3.24 (in part) SOME TYPICAL VESSEL SUPPORTS

3.25 ATTACHED STRUCTURES AND EQUIPMENT

3.25.1 Structures—General

Internal and external non-pressure structures and fittings attached to the vessel shall be designed in accordance with good engineering practice and shall be arranged as far as practicable without imposing local concentrated loads on the vessel wall. Loads from attached structures, equipment and fittings, should be carried by means of suitable stiffeners and/or spacers, directly to the vessel supports and thus to the foundations without stressing the vessel walls or ends. Where this is not practicable, such loads shall be supported in accordance with Clause 3.24. For attachments to transportable vessels see Clause 3.26.14.

Lugs, rings, brackets, and the like, should be designed so as to drain water away from, rather than towards, any insulation attached to the vessel. Pockets or crevices which could trap liquid and cause corrosion should be avoided.

3.25.2 Internal structures

Internal structures shall be designed to prevent failing in service. They should rest on top of their supports in preference to being suspended from them. Such structures and supports shall be constructed of material corrosion-resistant to their environment, or provided with additional metal where corrosion is expected. For structures which can be readily replaced the corrosion allowance need not be the same as for the vessel.

3.25.3 General method of attachment

Lugs, clips or supports for structures, lining, insulation, operating equipment and piping may be attached to the inside or outside of the vessel provided that allowance has been made to prevent excessive stresses or distortion in the vessel wall under all service conditions. Lugs, clips and supports welded to the vessel wall shall be of sufficient size to prevent over-stressing. They should be not more than twice the wall thickness.

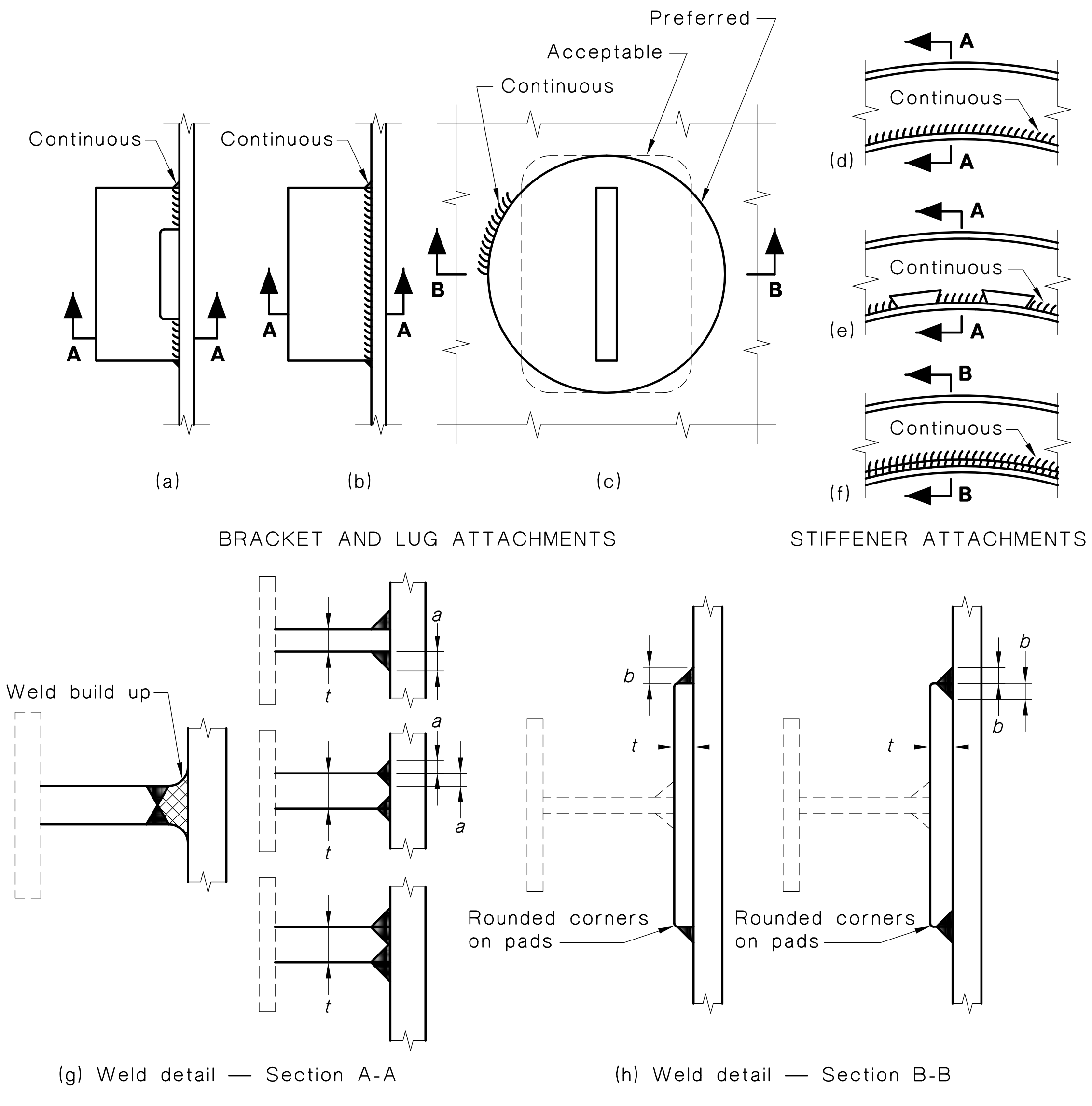
Resistance welded studs may only be used for non-pressure attachments to pressure parts and by agreement with the parties concerned.

Welded attachments shall be designed in accordance with Clause 3.5 and Figures 3.25(A) and (B), with attachment weld strength determined in accordance with Clause 3.19.3.5. Where practicable, all welds, particularly to pressure parts, shall be continuous. See AS 4458 for welding of attachments.

For clad construction when attachments are made to cladding and not directly to the base metal, it shall be demonstrated that the bond between the cladding and the base metal is adequate for the loads and complies with other relevant requirements of this Standard.

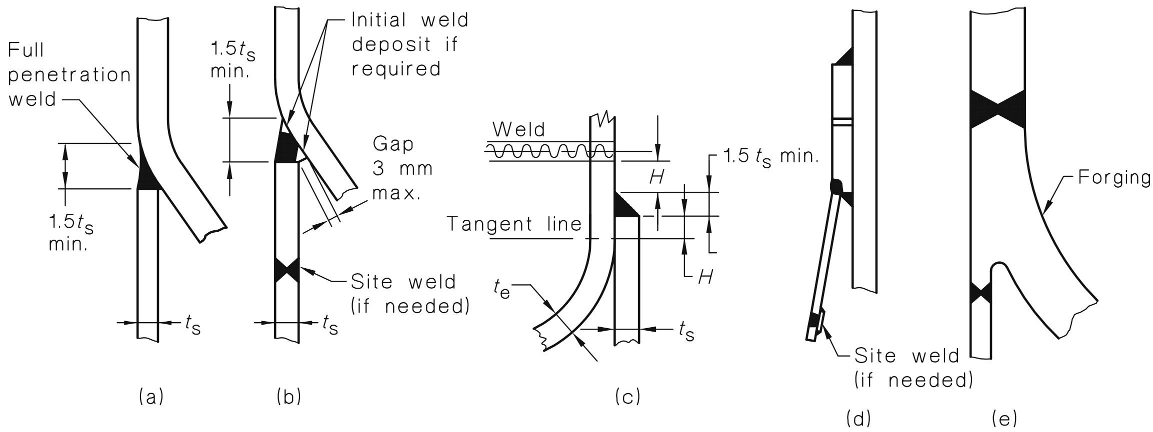
If a doubler plate is used and it either covers a vessel butt weld or the vessel will be subject to post weld heat treatment, the doubler plate should be provided with at least one tell-tale hole.

NOTE: There might be other reasons to include a tell-tale hole, such as to act as a breather during welding or to aid in the detection of possible cracking if the vessel is subject to cyclic thermal operation.



NOTE: $a \geq t/4$; $b \geq t/2$ where t = thickness of attached member.

FIGURE 3.25(A) BRACKET, LUG AND STIFFENER ATTACHMENT



NOTE: H ≥40 mm or $3t_e$ (whichever is the lesser).

FIGURE 3.25(B) SOME TYPICAL SKIRT SUPPORT ATTACHMENT

3.26 TRANSPORTABLE VESSELS

3.26.1 General

The design of transportable vessels, within the limits of Clause 1.3, shall comply with the requirements of this Standard and the additional requirements given in this Clause (3.26). The design shall also satisfy such additional requirements as are imposed by the relevant equipment or application Standards and the purchaser. (See Appendix E.)

NOTE: This Clause is concerned only with those basic requirements of a transportable vessel which pertain to its performance as a pressure vessel and includes its immediate attachments. For those particular requirements relating to its performance as a vehicle, see the relevant application Standards (e.g. AS 2809.1, AS 2809.3, AS 2809.4 and AS 2809.6). For guidance, see also ISO 20421 and ISO 21029 for transportable vacuum-insulated cryogenic vessels.

3.26.2 Types and application

Transportable vessels are pressure vessels designed for the transport of fluids under pressure and for the purpose of this Standard include the following types:

(a) *Road tanker vessels* These are vessels designed to be permanently mounted on or form an integral part of a road vehicle.

(b) *Rail tanker vessels* These are vessels that form part of a rail tank wagon, and are designed to be permanently mounted on an underframe on bogies or alternatively the vessel itself may form a structural part of the wagon.

(c) *Portable vessels* These are vessels that are designed in such a manner as to permit them to be moved (normally by road or rail) to various locations. Such vessels also include those which are fitted with suitable steel 'runners' or 'skids' and are generally known as 'skid or demountable tanks'.

(d) *Tank shipping containers* These are vessels also designed, manufactured, tested and inspected in accordance with AS 3711.6 and contained in a standard frame for multi-modal transport (sea, rail and road).

NOTE: For international transport, the IMDG Code applies.

Vessels that are used for the transport of material under no pressure but which are subject to pressure on discharge of contents may be regarded as static vessels, except that the design and manufacture of supports and attachments to the pressure parts shall comply with the requirements for transportable vessels.

3.26.3 General design

3.26.3.1 *Class of construction*

Transportable vessels shall be constructed to Class 1 requirements, with design strengths based on $R_m/4$, except as follows:

(a) Class 1 construction shall be used for uninsulated transportable vessels with flammable gases provided they are designed for fire resistance, equivalent to requirements of Paragraph L3.13(f). See Clause 2.5.3.2 for service limits related to low melting point materials.

(b) Class 2A construction may be used for—

  (i) portable, LP Gas vessels not exceeding 8000 L capacity; or

  (ii) transportable vessels with non-lethal fluid and with pressure times volume not exceeding 10 times that permitted for transportable vessels of hazard level C to AS 4343.

(c) Class 2A and 2B construction may be used for transportable vessels not exceeding 5000 L capacity with non-harmful fluid (to AS 4343).

(d) Class 3 construction may be used for vessels within the following limits on contents and pV:

  (i) Air and other non-harmful gases with $pV \leq 150$ MPa.L (e.g. vessels such as air receivers on road marking vehicles, braking and air-start vessels).

  (ii) Non-harmful liquids with $pV \leq 1000$ MPa.L (e.g. water vessels).

(e) Class 1H or 1S construction with design strengths to Class 1H or 1S criteria (see Appendix A)—

  (i) comply with Appendix M (Fatigue);

  (ii) are designed against the effect of collision (see Appendix N); and

  (iii) are designed for fire resistance equivalent to the requirements of Paragraph L3.13(f).

3.26.3.2 *Design pressure*

Single-wall uninsulated transportable vessels and associated pressure parts shall have a design pressure not less than that specified in the application Standard and where this is not specified the greater value of—

(a) 700 kPa; and

(b) the vapour pressure of the fluid at the maximum fluid service temperature determined from AS 2872 (or if desired, the vapour pressure at 50°C for vessels with a capacity greater than 500 L and at 46°C for vessels with capacity greater than 2000 L).

For fully insulated transportable vessels with substantial external protection the design pressure shall be not less than the greater of—

(i) 170 kPa; and

(ii) the vapour pressure at the maximum service temperature of the fluid, normally determined by the set pressure of the pressure relief device.

3.26.3.3 *Openings*

No openings shall be provided on the outside circumference of the upper 2/3 of the cylindrical shell (i.e. 120° either side of the top of the vessel), unless provided with a recess. The recess shall ensure that all pressure relief valves and other fittings lie within the outline of the cylindrical shell as protection from damage in a roll over.

Vessels for chlorine or more toxic substances shall have only one manway. The manway opening and closures to openings shall lie within the outside vessel envelope.

3.26.3.4 *Loadings*

Transportable vessels, supports and attachments shall be designed to withstand the loadings in accordance with Clause 3.2.3 and the following:

(a) *Road tanker vessels*—loadings in any direction equal to twice the force due to the mass of the vessel together with its attachments and contents, when filled to the maximum permissible loading and also the loadings in Clause 3.26.3.7.

(b) *Rail tanker vessels*—loadings due to shunting, and rail service as specified by the rail authority.

(c) *Portable vessels (excluding skid tanks)*—loadings in any direction equal to twice the force due to the mass of the vessel together with its attachments and contents, when filled to the maximum permissible loadings and also loadings in Clause 3.26.3.7.

(d) *Skid tanks*—loadings in any direction equal to four times the force due to the mass of the vessel together with its attachments and contents, when filled to the maximum permissible loading.

(e) *Shipping containers*—loadings specified in AS 3711.6.

A1 (f) *Class 1H, 2H, 1S and 2S vessels*—fatigue loads and cycles to be agreed between the designer, purchaser/owner and design verifier (see Appendix M).

3.26.3.5 *Structural integrity*

This Clause covers stresses which act over the entire cross section of the vessel and thus specifically excludes local stresses which are covered in Clause 3.24.

The maximum membrane stress calculated at any point in the vessel under this Clause shall not exceed that listed for the material in Clause 3.3.1.1. Alternatively test or analytical methods or a combination thereof, may be used if the methods are accurate and verifiable.

Corrosion allowance shall be added to the design minimum thickness.

A2 **3.26.3.6** *Design by calculation or finite element analysis*

Design shall be conducted by calculation or finite element analysis as follows:

(a) *Calculation*

If the design is conducted by calculation, the calculation shall allow for the combined effect of pressure loadings (both circumferential and longitudinal stresses), torsion, shear, bending and acceleration of the vessel as a whole (both forward and rearward). Consideration shall be given to the effects of thermal gradients and fatigue.

The vessel design shall include calculation of membrane stresses generated by design pressure, the weight of contents, the weight of structure supported by the vessel wall, the loadings specified in Clauses 3.26.3.4 and 3.26.3.7 and the effect of temperature gradients resulting from vessel contents and ambient temperature extremes. When dissimilar materials are used, their thermal coefficients shall be used in calculation of thermal stresses. See Clause 3.26.10.1 for stresses that occur at pads, cradles or other supports.

(b) *Finite element analysis*

If a transportable vessel is designed using finite element analysis the resulting design shall comply with the static strength requirements of Appendices H and I as appropriate, using the static load cases given in Table 3.26.3.8.1 and shall, regardless of class, comply with the fatigue requirements of Appendix M. The cyclic loading shall be the value agreed between the designer and operator or, where there is no agreement, the value given in Table 3.26.3.8.2.

3.26.3.7 *Combined stresses*

The equivalent stress from static or dynamic loads listed below, or a combination thereof that can occur at the same time is given by the following equations:

$$S_E = \frac{(S_c + S_l)}{2} \pm \sqrt{\left(\frac{S_c - S_l}{2}\right) + S_s^2} \qquad \ldots 3.26.3.7(1)$$

where

$$S_l = S_c/2 \pm S_b \text{ (longitudinal stress)} \qquad \ldots 3.26.3.7(2)$$

$$S_s = 2Q/\pi\, D_m^2 t \text{ (shear stress)} \qquad \ldots 3.26.3.7(3)$$

and the value of—

$$S_E,\ S_c \text{ and } S_l \leq \eta f \qquad \ldots 3.26.3.7(4)$$

When S_c and/or S_l are compressive (negative), these shall be limited as per Clause 3.7.5.

For the purpose of this Clause, refer to Clauses 3.7.2 and 3.7.5 for notation and equations except that the force due to the mass shall be incorporated in the bending moment, *M*.

The equivalent stress shall be evaluated at any point for the following loading conditions:

(a) The circumferential stress generated by internal or external pressure (or both).

(b) The longitudinal tensile stress generated by internal pressure.

(c) The tensile or compressive stress generated by the axial load resulting from a decelerative force equal to twice the static weight of the fully loaded vessel applied independently to suspension assembly at the road surface.

(d) The tensile or compressive stress generated by the bending moment resulting from a decelerative force equal to twice the static weight of the fully loaded vessel applied independently to each suspension assembly at the road surface.

For vessels with internal baffles, the decelerative force may be reduced by '0.25*g*' for each baffle assembly, but in no case may the total reduction in decelerative force exceed '1*g*'.

(e) The tensile or compressive stress generated by the axial load resulting from an accelerative force equal to the static weight of the fully loaded vessel applied to the horizontal pivot of the fifth wheel supporting the vessel, if used.

(f) The tensile or compressive stress generated by the bending moment resulting from an accelerative force equal to the static weight of the fully loaded vessel applied to the horizontal pivot of the fifth wheel supporting the vessel, if used.

(g) The tensile or compressive stress generated by the bending moment produced by a vertical force equal to three times the static weight of the fully loaded vessel.

(h) The shear stress generated by a vertical force equal to three times the static weight of the vessel and contents.

(i) The lateral shear stress generated by a lateral accelerative force which will produce an overturn but not less than 0.75 times the static weight of the fully loaded vessel, applied at the road surface.

(j) The torsional shear stress generated by a lateral accelerative force which will produce an overturn but not less than 0.75 times the static weight of the fully loaded vessel, applied at the road surface.

(k) Peak stresses for Class 1H and 2H vessels shall comply with Appendix M.

A2

3.26.3.8 *Loads for use in the finite element analysis design of transportable vessels*

3.26.3.8.1 *Static strength analysis*

The vessel (inner and outer vessels if double-walled) and supports shall be designed for the quasi-static forces associated with the combinations of dead weight and acceleration given in Table 3.26.3.8.1. The masses considered shall include the vessel(s), maximum permissible content mass, supports, piping, insulation and any other item supported by the vessel. Each load case shall be considered separately, but all component loads within a given load case shall be applied simultaneously, including the internal and/or external design pressure.

If a vessel design is required to meet the provisions of an additional design code, such as IMDG, the specific load cases provided in that code will also apply.

TABLE 3.26.3.8.1

FINITE ELEMENT ANALYSIS DESIGN LOADS FOR STATIC STRENGTH ASSESSMENT

Transportable vessel type	Load case	Dead load and 'g' load factors				
		Forward (see Note 2)	Backward (see Note 3)	Down (see Note 4)	Lateral	Dead load down
Road tankers, shipping containers and portable tanks (excluding skid tanks)	1	2.0				1.0
	2		2.0			1.0
	3			2.0		1.0
	4				2.0	1.0
Rail tankers with cushioning devices (see Note 1)	1	2.0				1.0
	2		2.0			1.0
	3			1.0		1.0
	4				2.0	1.0
Rail tankers without cushioning devices	1	4.0				1.0
	2		4.0			1.0
	3			1.0		1.0
	4				2.0	1.0
Skid tanks	1	4.0				1.0
	2		4.0			1.0
	3			3.0		1.0
	4				4.0	1.0

NOTES:

1 The cushioning devices should be tested to demonstrate their ability to limit forces transmitted from the coupler to the tank is less than twice the weight of the tank filled to its rated capacity at a 16 km/h impact.

2 The forward 'g' load factor models a deceleration (e.g. during braking), that generates a forward-directed inertia force resulting for example in an increase in the hydrostatic pressure (of contained liquid in the vessel) in the leading end of the vessel compared to a decrease of same at the trailing end of the vessel. In the case of articulated road tankers, the required force shall be considered as being applied through the king pin. With respect to the provision of baffles to attenuate and distribute the dynamic fluid forces of contained liquid, each compartment between baffles can be treated in the FEA as a sealed compartment with respect to the hydrostatic pressure generated by forward and rearward acceleration.

A2 3 The backward ‘g’ load factor models a forward acceleration (that generates a resisting backward-directed inertia force) resulting for example in an increase in the hydrostatic pressure (of contained liquid in the vessel) in the trailing end of the vessel compared to a decrease of same at the leading end of the vessel. In the case of articulated road tankers the required force shall be considered as being applied through the king pin.

4 The downward ‘g’ load factor models an upward acceleration (that generates a downward-directed inertia force) resulting for example in an upwards force of three times the loaded weight of the vehicle being applied from the road to the tyres of a road tanker, given a downward load factor of 2 combined with the dead load (load case 3).

3.26.3.8.2 *Fatigue strength analysis*

The fatigue strength of the vessel or component shall be determined using linear elastic finite element analysis in accordance with Appendix M. The fatigue load cases and number of cycles are the subject of agreement between the designer and operator. However, in the absence of such information, the load cases and number of cycles given in Table 3.26.3.8.2 shall be used for the relevant transportation mode.

For a given transportable vessel type, the peak stress range shall be calculated for the simultaneous application of the three component acceleration loads given in Table 3.26.3.8.2. These loads may be applied as accelerations through the loaded structure’s centre of gravity, with the vessel restrained appropriately at its support points (e.g. lateral and vertical fixation at wheels and kingpin for the vertical and lateral components, and kingpin only for the axial component of acceleration).

For Classes 1H and 1S the fatigue damage shall also be assessed for an agreed number of design pressure cycles. The cumulative damage (Miner’s summation) for the combined pressure and transportation cyclic loading shall not exceed 1.0.

TABLE 3.26.3.8.2

FINITE ELEMENT ANALYSIS DESIGN LOADS FOR FATIGUE ASSESSMENT

Transportable vessel type	‘g’ load factors (total range)			Number of cycles
	Axial	Vertical	Lateral	
Road tankers, shipping containers and portable tanks (including skid tanks)	1.4	3.0	1.4	10^4
Rail tankers with cushioning devices (see Note)	2.0	3.0	1.4	10^4
Rail tankers without cushioning devices	4.0	3.0	1.4	10^4

NOTE: The cushioning devices should be tested to demonstrate their ability to limit forces transmitted from the coupler to the tank is less than twice the weight of the tank filled to its rated capacity at a 16 km/h impact.

3.26.4 Materials

3.26.4.1 *General*

For materials see Clause 2.5 for general limits and Clause 2.6 for provision against brittle fracture. All pressure-retaining materials shall be suitably ductile, with $A_5 \geq 14\%$.

Material for pads shall be the same material group, as for the vessel to which they are attached.

Where low melting point materials are used for flammable products the following conditions shall be met:

(a) The vessel, including manholes and nozzles, shall be insulated with a material agreed between the parties concerned.

(b) The applied insulation shall have a thermal conductance at 900°C temperature difference of not more than 0.43 W/m^2K.

(c) The entire insulation shall be covered with a weather-tight steel jacket at least 3 mm thickness.

(d) The interior surface of the jacket shall be corrosion resistant or protected against corrosion.

3.26.4.2 *Material for vessels with lethal and very toxic (e.g. chlorine, sulfur dioxide and ammonia) contents*

Chlorine vessels shall be constructed of carbon-manganese fine grain steel plate with impact test requirements in both the longitudinal and transverse direction at a temperature of minus 40°C and energy values to be met shall be in accordance with Table 2.6.2.

Sulfur dioxide vessels shall be constructed of carbon-manganese fine grain steel.

A vessel in anhydrous ammonia service shall be constructed of steel. The use of copper, silver, zinc or their alloys is prohibited. Baffles made from aluminium may be used only if joined to the tank by a process not requiring postweld heat treatment of the tank. Compliance with AS/NZS 2022 is required.

3.26.4.3 *Minimum thickness*

For vessels not covered by other Standards or not suitably safeguarded, the thickness of the shell and ends (or the combined wall thickness of jacket and the shell or ends) t_{min} shall be not less than—

(a) 5 mm;

(b) that required in Clause 3.4.3; and

(c) for shell and ends over 1000 mm diameter in contact with the pressurized contents not less than that given by Equation 3.26.4.3:

$$t_{min} = \frac{C}{\left(R_m \times A_5\right)^{\frac{1}{3}}} \qquad \text{. . . 3.26.4.3}$$

where

A_5 = guaranteed minimum percentage elongation (see Paragraph A2 for definition)

C = 107 for vessels ≤1.8 m diameter

= 128 for vessels >1.8 m diameter

= 64 for vacuum insulated vessels ≤1.8 m diameter

= 85 for vacuum insulated vessels >1.8 m diameter

R_m = ‘guaranteed’ minimum tensile strength at design temperature, in megapascals

t_{min} = minimum thickness of shell and ends at design temperature, in millimetres

For vessels with the wall thickness less than 10 mm, circumferential reinforcement shall be assessed in accordance with Clause 3.9 for full vacuum design.

For vessels with lethal fluids, chlorine and sulfur dioxide, the minimum (or combined) wall thickness including corrosion allowance shall be at least 15 mm.

3.26.5 Corrosion allowance

A vessel subject to thinning by corrosion, erosion or mechanical abrasion due to the contents shall be protected by providing the vessel with a suitable increase in material thickness (see Clause 3.2.4).

For chlorine and sulfur dioxide vessels corrosion allowance of 20% of minimum calculated thickness or 2.5 mm, whichever is less, shall be provided.

3.26.6 Welds

Welded longitudinal joints in the shell shall be located in the upper half of the vessel. No welding, other than that of attachments to pads provided for that purpose, shall be performed on the vessel after final heat treatment.

3.26.7 Heat treatment

All transportable vessels except those constructed from high alloy steels or non-ferrous materials or of Class 2 construction shall be postweld heat treated in accordance with AS 4458.

3.26.8 Protection against damage

The design, manufacture or installation (or both) of transportable vessels shall minimize the possibility of damage to pressure-retaining parts due to collision, overturning and fire.

The strength of the attachment of appurtenances to shells, ends or mounting pads, shall be such that when force is applied to the appurtenance (as mounted and normally equipped) in any direction except normal to the vessel shell or within 45° thereof, the attachment shall fail completely without such damage occurring to the shell or end as would affect the product-retention integrity of the vessel.

All structural and non-pressure attachments to the shell, ends, or mounting pads, such as stiffening rings, lifting lugs, baffles, rear bumper and overturn protection shall comply with the requirements of this Clause (3.26).

All closures for filling, emptying, inspection or other openings shall be protected by enclosing these fittings within the outline equal to the diameter of the vessel or within a dome attached to the vessel or by guards (see Clause 3.26.12), or by suitable emergency internal valves and a shear section outboard of the valve. Manholes in the ends of a vessel need not meet these requirements.

If f exceeds $R_m/3.0$, the vessel shall comply with Paragraph L3.13(f).

3.26.9 Stability and clearances

The stability and clearances of a road or rail tanker shall be adequate to ensure safe transport and shall comply with the requirements of the appropriate authorities concerned (e.g. motor transport authorities and railway departments). The minimum road clearance of the vessel or a protective guard located between any two adjacent axles on a vehicle shall comply with AS 2809.1 and the Australian Design Rules for vehicles.

Ground clearance for portable vessels shall be at least 50 mm.

3.26.10 Vessel supports

3.26.10.1 *General*

In addition to meeting the applicable requirements of Clauses 3.24 and 3.25 supports for transportable vessels shall be designed to resist the appropriate forces (see Clauses 3.2.3 and 3.26.3.4) and meet the following requirements:

(a) Vessels which constitute in whole or in part the stress-members used in lieu of a frame shall be supported by external saddles subtending at least 120° of the shell circumference continuously or by other means demonstrated to have equivalent impact and fatigue performances. Alternatively the design shall satisfy Clause 3.1.3.

(b) Local stresses developed in the shell or ends at saddles due to shear, bending and torsion shall be calculated in accordance with the recommended methods in Appendix N, or alternatively the design shall satisfy Clause 3.1.3. Stresses shall be limited to those given in Clause 3.3.1.

(c) Vessels that are not constructed integrally with or not welded directly to the frame of the vehicle shall be provided with turn-buckles or similar positive devices for the drawing the vessel tightly to the frame. In addition, suitable anchors or stops shall be attached to the frame or the vessel (or both) to prevent relative motion between them due to stopping, starting or change of direction. Devices for drawing-down the vessel shall secure the vessel to the vehicle in a safe manner that will not introduce excessive stress in the vessel.

(d) Stresses developed at components of supports, anchors, stops and the like shall be calculated and limited to those given in Clause 3.3.1.

(e) The ‘g’ loadings shall be applied through the vessel centre line and shall be assumed to be uniformly distributed.

3.26.10.2 *Pads*

Vessel supports and anchor points permanently attached to the vessel wall (see Clause 3.26.3.5) should be attached by means of pads which should be of the same material as the vessel wall.

Pads shall—

(a) have thickness not exceeding 1.5 times the thickness of the shell or head and not less than 5 mm (the throat thickness of the fillet weld joining pad to vessel shell shall be not more than the shell thickness);

(b) extend at least 4 times its thickness in each direction beyond the toe of the weld attaching the support;

(c) have its corner rounded to a radius equal to at least four times the thickness of the pad;

(d) have weep or tell-tale holes, drilled or punched before attachment to the vessel and subsequently filled to prevent ingress of moisture, in accordance with Clause 3.19.3.4;

(e) be attached to the vessel by continuous fillet welding;

(f) be designed so that firstly the attachment of the appurtenance to the pad, and secondly the pad, will fail completely without damage to the shell or end; and

(g) be located clear of the main joints of the vessel (see Clause 3.5.1.3).

3.26.10.3 *Lugs*

Support, holding down or lifting lugs may be integral with the doubling or mounting pads on portable vessels.

3.26.11 Rear impact protection

Each tanker shall be provided with a system of bumpers or barriers (or both) to protect the vessel from rear impact and underrun in accordance with AS 2809.1.

3.26.12 Guards for vessel fittings

When guards are required to protect vessel fittings from damage, which would result in leakage of the contents in the event of over-turning of the vehicle, they shall be designed and installed to withstand a vertical force equal to twice the force due to the mass of the loaded vessel without allowing application of force to the fitting, and a horizontal force in any direction of half the force due to the mass of the loaded vessel. These design forces may be considered individually. The maximum calculated stress shall not exceed 75% of the tensile strength of the material.

3.26.13 Lifting lugs

Lifting lugs or eyes for a portable vessel shall be designed to permit safe lifting of the vessel. Each lug of a portable vessel shall be designed to withstand a static force in any direction equal to twice the force due to the mass of the vessel and its contents.

3.26.14 Attachment of structures

The attachment of structures to the walls of transportable vessels shall meet the requirements of Clause 3.25 and other requirements of this Clause 3.26. They shall be designed to avoid fatigue.

Lightweight attachments such as a conduit clip, brakeline clip or placard holder shall—

(a) be constructed of a material of lesser strength than the vessel wall material;

(b) not be more than 70% of the thickness of the material to which it is attached.

The attachment may be secured directly to the vessel wall if the device is designed and installed in such a manner that, if damaged, it will not affect the lading retention integrity of the tank.

Major load-carrying structures such as supports, lifting lugs, bumpers, baffles and surgeplates and those structures which may induce fatigue forces into the shell, shall be attached by pads and shall meet the other requirements of Clause 3.25 and this Clause 3.26.

3.26.15 Pressure relief valves

Pressure relief valves shall be sized in accordance with Section 8 with particular reference to the integrity of insulation in accidents and under fire conditions (see Clause 8.2.2.2) and shall comply with the relevant application Standard. Arrangements shall be made to protect the relief valves from damage caused by the vessel overturning.

3.27 QUICK-ACTUATING CLOSURES

3.27.1 Types of closures

Quick-actuating closures also designated as quick-opening closures, are closures designed for more rapid opening and closing than closures of the multi-bolted type. Most quick-actuating closures are within the following classifications:

(a) Interlocking lug-type closures, with load-bearing lugs on the door engaging/disengaging with lugs on the shell or a ring on the shell by part rotation of one set of lugs.

(b) Expanding or contracting ring-type closures, where the holding is by a ring connecting the door to the vessel.

(c) Clamp-latch-type closures, using multiple clamps to connect the door to a flange on the vessel. May incorporate ring-type closures, see Item (b).

(d) Bar-locking-type closures, where multiple bars are actuated simultaneously to connect the door to the vessel.

(e) Single swing bolt-type, beam and yoke closures, where the closure is retained by a bridge or equivalent across the closure with a retaining device such as a latch or swing bolt. The closure door may function as the beam.

(f) Sliding-type closures, where the door is retained by guides and opens in the same plane as the vessel opening face.

NOTE: Internally fitted doors of the type covered by Clause 3.15.5 are not covered by this Clause (3.27).

3.27.2 Design requirements

Many types of quick-actuating closures are available and it is not practicable to write detailed requirements for each or to prevent the circumvention of safety devices. Where such closures are used, they and their appurtenances shall be designed and installed to comply with the following requirements, with the exception of swing bolt closures of Clause 3.27.3:

(a) The holding elements (see Note) shall be designed so that failure of any one holding element cannot result in the release or failure of all the other holding elements. A single bridge or yoke-type holding element with external closure may be used for openings having an internal cross-sectional area not exceeding 0.25 m^2.

NOTE: Holding elements are those elements which hold the closure to the vessel and resist the end pressure and closure force, e.g. lugs, levers and bridges.

(b) Locking elements shall be provided where applicable (see Note).

NOTE: Locking elements are elements that lock the holding elements firmly in position while the vessel is under pressure.

(c) The locking arrangement shall be designed so that failure of any one of the locking elements does not result in the release of the closure under pressure.

(d) The closure shall be arranged so that it may be determined by visual external inspection that the holding elements are in good condition and that their locking elements, when the closure is in the closed position, are in full engagement.

(e) The closure, its holding elements, and locking elements shall be of a type that requires all parts to be fully engaged before pressure can be built up in the vessel.

(f) Pressure tending to force the closure clear of the vessel shall be reduced to a low value (see Note) before the holding elements can be operated, and shall be reduced to atmospheric pressure before the closure can be fully opened.

The closure shall not open suddenly in one stroke. The initial opening action shall be such as to ensure that there is sufficient time to allow all pressure to be released before the holding elements are fully disengaged, e.g. by initially developing a gap of approximately 3 mm between the closure and the vessel, or by other equivalent means as appropriate for the type of closure.

NOTE: The following Equation may be used to estimate the value to which the pressure should be reduced before disengagement of the holding elements can be started:

$$P = \frac{1.30\times10^6}{D^2}+7$$

where

P = maximum pressure in vessel before holding elements can be operated, in kilopascals

D = inside diameter of opening, in millimetres

(g) Where compliance with Items (e) and (f) is not inherent in the design of the closure and its holding elements, provision shall be made so that devices to accomplish this can be added when the vessel is installed. Pressure gauges and pressure switches shall not suffice for this requirement.

(h) A closure with several holding elements shall have locking elements dimensioned and fitted so that during operation individual holding elements are uniformly engaged.

(i) Holding and locking elements shall be able to withstand all forces and moments encountered during closure and subsequent operations.

(j) The design strength for elements shall be the same as permitted for the design of flanged joints (see Clause 3.21).

(k) Hinges, holding and locking elements and other parts shall have adequate allowance for corrosion and wear.

(l) Seals and gaskets shall provide leaktightness acceptable to the parties concerned when the cover is in the closed and locked position. For vessels containing lethal or flammable fluids, leaktightness shall be ensured by the adoption of appropriate special measures.

Seals and gaskets shall be positively separated from the closure faces before the holding elements can be fully released. Where adherence of contents or other matter retards free disengagement of the seal, positive mechanical means shall break the seal prior to full release of holding element(s).

(m) When installed, vessels having quick-actuating closures shall be provided with a pressure-indicating device visible to the operator from the closure-operating area. Additional warning devices, whether audible or visible, are not prohibited, but shall not be considered as satisfying any of the foregoing requirements.

(n) Manually operated closures shall be provided with an audible or a visible warning device that shall set off an alarm to the operator if an attempt is made to pressurize the vessel when the holding elements are not fully engaged, or if any attempt is made to disengage the locking elements while the vessel is pressurized.

(o) For automatically operated closures, the operating, control and safety system shall provide reliable and safe closing, locking and opening of closures as required by this Clause and Clause 8.2.7.

(p) Switchable guarding shall be provided where necessary, to control risks in movement of closure.

For guidance on machine guarding systems, refer to AS 4024.1.

Surfaces that are to be handled during the opening operation should have a temperature not exceeding 55°C for metal or 60°C for heat-insulating material.

For Lethal Gases, or Very Harmful Gases, provision should be made for the gas to be purged prior to release of the seal.

3.27.3 Swing bolt closures

Where covers with slots (or equivalent) are engaged by bolts that are released by pivoting clear of the cover, the following requirements apply:

(a) The design shall prevent inadvertent or immediate release of the cover after loosening of any bolt.

(b) Single, two-bolt, and three-bolt closures shall—

  (i) have pressurized areas not exceeding 0.125 square meters; and

  (ii) be used only for vessels of hazard levels D or E (determined according to AS 4343).

(c) A minimum of four swing bolts shall be used on vessels that—

  (i) contain lethal contents;

  (ii) contain very harmful contents; or

  (iii) have a hazard level A, B or C according to AS 4343.

(d) Pins on which the bolt swings shall be supported at each end, and be securely retained in the carrying lugs.

(e) Bending, shear and bearing stresses in the pins shall comply with Clause 3.3.

(f) Washers, if provided, shall have minimum thickness of 0.25 × nominal bolt diameter.

(g) Each nut (or washer if fitted), shall seat into a recess or behind a barrier on the cover. The recess or barrier shall prevent swinging of the bolt until the nut has been loosened by the greater of 3 mm or 0.2 × nominal bolt diameter.

3.28 METALLIC EXPANSION JOINTS

Metallic expansion joints shall comply with the requirements specified in the Standards of the Expansion Joint Manufacturers Association (EJMA), ASME BPV-VIII, EN 13445, or equivalent Standard agreed by the parties concerned. Design strengths (primary general membrane stress) shall not exceed limits in this Standard.

3.29 PRESSURE VESSELS FOR HUMAN OCCUPANCY

3.29.1 Static service

Static vessels for human occupancy shall comply with the following requirements:

(a) The requirements for Class 1 vessels set out in this Standard.

(b) Welded joints of Type A, B and C (see Figure 3.5.1.1) shall be double-welded butt joints without retained backing strips (as set out in Table 3.5.1.7), except as set out in Item (c).

(c) An intermediate end attached in accordance with Figure 3.12.6(k) may be used subject to the following additional requirements:

   (i) The allowable working pressure shall not exceed 0.95 MPa.

   (ii) The stress in the two connecting shells and the intermediate end shall not exceed $0.7 \times f$.

   (iii) The straight flange of the intermediate end shall be at least 38 mm long.

   (iv) The minimum leg length for the fillet weld (attaching the straight flange to the shell) shall be the lesser of $0.5 \times t$ or 6 mm.

   (v) The shear stress in the butt weld and the fillet weld shall not exceed $0.2 \times f$ (where f is for the shell material.)

   (vi) The butt weld (joining the shells) shall undergo 100% NDE.

(d) Non-destructive examination of joints shall comply with Class 1H requirements (except as required in Item (c)(vi)).

(e) The design of any viewports and windows shall comply with the relevant Clauses of ASME PVHO-1.

(f) The vessel shall be marked 'AS 1210 Class 1-PVHO'.

   NOTE: The requirements for pressure vessels for human occupancy as set out in AS 4774.2, might also apply.

3.29.2 Non-static and general service

For marine and transportable applications, or where the installation environment is unpredictable, it is recommended that vessels should comply with the design, materials, manufacturing, testing and inspection requirements of ASME PVHO-1.

3.30 BURIED AND MOUNDED STORAGE VESSELS

3.30.1 Design conditions

Buried and mounded storage vessels shall be designed and manufactured to Class 1 or 1H construction. The design pressure shall be at least the maximum pressure expected under all operating conditions and not less than that specified in the pressure equipment application Standard, e.g. AS/NZS 1596. The maximum design temperature shall be no less than 50°C. The minimum design temperature shall be the lower of 0°C, the minimum operating temperature or the temperature at which the contents are received. The fatigue life shall be based on a minimum of 1000 cycles.

3.30.2 Vessel support

Buried or mounded storage vessels may be supported on saddles, or on a continuous sand bed supporting the vessel over its full length on a subsoil with little uneven settlement which may cause problems with connected piping and exert considerable stresses in the vessel wall.

3.30.3 Design loads

Buried or mounded storage vessels will have additional loads caused by the mound and the method of support.

The earth mound and supports will cause bending moments, normal forces and shear forces in the vessel wall which may not be carried by the relatively thin vessel wall, especially in large diameter vessels. Then continuous internal stiffening rings should be considered.

The design of the vessel shall consider the effect of the following loads, and combinations thereof as applicable:

(a) The dead weight of vessel, contents and the effective earth mound assumed to be supported on top of the vessel.

(b) Live loads by machinery and equipment during installation, maintenance or during operation.

(c) Soil pressures and friction loads during expansion and contraction of the vessel and attachments caused by temperature and pressure variations.

NOTE: Special attention should be given to the non-uniform nature of the external soil pressures which will result in circumferential bending moments in the vessel wall.

(d) Local loads in vessel wall at nozzles and attachments.

(e) Buoyancy forces. Special care shall be given to hold down bracket attachment to vessel with loads due to buoyancy of the empty vessel in fully flooded site condition.

(f) For mounded vessels on supports, earthquake loads for the vessel and contents in horizontal and vertical accelerations at the centre of the vessel with no support assumed from the mound to the vessel.

The design of the vessel shall consider the effect of the loads described above for the fully corroded vessel under full design conditions including vacuum if required by design, during shutdown and maintenance with the vessel empty and at atmospheric pressure, and during site hydrostatic testing if required.

3.30.4 Material

The specified minimum tensile strength of the vessel shell shall not be greater than 510 MPa.

3.30.5 Pipe connections

Buried vessels shall have pipe connections grouped together at the top of the vessel, preferably through a tower manway cover. Piping connections shall be arranged to facilitate periodic inspection of safety valves and fittings. Valve pits shall be designed to either prevent the ingress of water, or to drain it away.

Mounded vessel may use grouped pipe connections located at one end of the vessel provided that connected pipelines are clear of restraints and infill material, e.g. by passing through ducting; and the vessel support nearest to the pipe connection end is fixed. The second support may need to be designed to accommodate axial movement due to expansion and contraction.

3.30.6 Nozzles

No screwed connections or flanged joints shall be used buried. Nozzles shall be of integral construction with full penetration weld (see Figures 3.19.9 and 3.19.3B(f) to (k) for examples).

3.30.7 Corrosion allowance

An internal and external corrosion allowance shall be assessed in accordance to Clause 3.2.4, but in no case less than 1.5 mm for the external surfaces of the vessel and its integral attachments.

3.30.8 Coating and cathodic protection systems

Notwithstanding provision of corrosion allowance, an adequate coating system and a cathodic protection system shall be provided and satisfactorily tested prior to the vessel installation.

3.31 VESSELS OF NON-CIRCULAR CROSS-SECTION

Vessels of rectangular or obround cross-section, or of circular cross-section with a single diametral stay plate, shall be designed by—

(a) the method described in AS 1228;

(b) the design method in Appendix 13 of ASME BPV-VIII-1;

(c) the design method in EN 13445;

(d) other design methods given in Clause 3.1.3.

Design strength shall be from this Standard (see Clause 2.3.3 for design strengths for structural grade stiffeners).

3.32 FIRED PRESSURE VESSELS

3.32.1 Scope and application

This Clause 3.32 specifies additional requirements for fired pressure vessels, i.e. vessels heated directly by fire, the products of combustion, electrical power or similar high temperature means such as focussed solar radiation or molten or high temperature metal, e.g. cooling rolls.

This Clause applies to fired heaters and includes fired process heaters, fired water heaters (i.e. water heated below 100°C), fired oil heaters, fired vaporizers, and similar fired vessels.

This Clause does not apply to boilers (i.e. heating water or steam at above 100°C), nor to sterilizers, domestic hot water heaters and other equipment excluded by AS/NZS 1200, fired vats or other containers with a large vent area direct to atmosphere.

NOTE: Water heaters and other vessels with small vent pipes have exploded when vents are blocked.

3.32.2 Construction Standards

3.32.2.1 *General*

Fired vessels shall be designed and manufactured in accordance with one of the following:

(a) The requirements of this Standard and the additional requirements in Clauses 3.32.3 to 3.32.6;

(b) AS 1228, or equivalent, with suitable agreed provisions to cater for the particular heated fluid, when properties are significantly different from water.

(c) An applicable alternative Standard set out in Clauses 3.32.2.2 to 3.32.2.6.

Marking with 'AS 1210' applies only where the design, manufacture and testing complies with Item (a) above.

3.32.2.2 *Water heaters*

Water heaters (including heaters for beverages and liquids predominantly of water) may alternatively comply with AS 1056.1, AS 4552 for gas types, AS 3500.4, AS/NZS 4692 (series), or equivalent Standard and capacity. Water heaters of hazard level E to AS 4343 or lower may be made to good engineering practice but not necessarily to the full requirements of this Standard, i.e. where any one of the following apply:

(a) Of any pressure or volume but design temperature is less than 65°C.

(b) Of any volume but design pressure is less than 0.05 MPa and design temperature is less than 100°C.

(c) Of any pressure but volume less than 10 L and design temperature less than 100°C.

(d) Of any pressure and volume if product of design pressure and volume is less than 300 000 MPa L, and design temperature less than 90°C.

3.32.2.3 *Electrically heated calorifiers*

Electrically heated calorifiers may alternatively comply with BS 853.

3.32.2.4 *Fired process heaters*

Fired process heaters may alternatively comply with industry Standards.

3.32.2.5 *Fired LP Gas vaporizers*

Fired LP Gas vaporizers may alternatively comply with AS/NZS 1596.

3.32.2.6 *Fired organic fluid and vaporizers*

Fired organic fluid and vaporizers may alternatively comply with AS 1228.

3.32.3 Design features

The design shall be in accordance with the relevant construction Standard and the following:

(a) For design temperature, see Clause 3.2.2.1.

(b) The class of construction shall be determined as for unfired vessels except for welded joints, which are specified in Clause 3.32.5.

(c) For ferritic steel, the minimum corrosion allowance shall be 0.75 mm.

(d) Suitable provision shall be made for each of the following:

  (i) To limit thermal stresses and distortion arising from local or general differential heating or thermal expansion of parts.

  (ii) To avoid thermal cracking due to high local stress.

(iii) To avoid thermal fatigue cracking due to cyclic temperature changes due to operational needs or unstable fluid flow.

(iv) To limit thickness of parts exposed to high radiant heat, e.g. 25 mm maximum for carbon steel.

3.32.4 Welded joints subjected to heating

All longitudinal and circumferential joints subject to radiant heat shall be full penetration double welded. When these joints are not subject to radiant heat they may be single-welded with backing strip. Other joints, e.g. at nozzles, stiffeners, attachments and supports, shall be full penetration when subject to radiant heat and preferably full penetration when not subject to radiant heat.

Tube attachments shall provide good thermal contact preferably with a weld throat thickness at least equal to the tube thickness. Incomplete penetration welds, slip-on, socket-welding and threaded joints shall be limited to metal temperatures not exceeding 250°C.

Postweld heat treatment is required where the thickness at welded joints exceeds one half that thickness requiring postweld heat treatment in AS 4458.

3.32.5 Safety controls and devices

Fired vessels shall be provided with pressure and temperature controls, energy input controls, level controls, flow controls, safety devices, valves and other fittings to—

(a) permit safe operation;

(b) effectively limit pressure to not more than 1.1 times the design pressure;

(c) limit the design temperature and fluid temperature to not more than the design value, except for short-term variation permitted by this Standard;

(d) control the risk of pressure explosion and electrical and fire hazards; and

(e) comply with the applicable requirements of Section 8.

3.32.6 Valves, gauges and other fittings

Suitable fittings shall be provided to permit safe operation of the vessel and to comply with the relevant requirements of Section 8. See also AS 2593 for general guidance on valves, gauges and other fittings.

Vent pipes (or equivalent arrangements) for vessels with a design pressure exceeding 0.05 MPa due solely to static head, shall comply with each of the following:

(a) For water heaters, have a vent size complying with Table 3.32.6.

(b) Be designed and sized to minimize fouling and prevent blockage or restriction which could cause excessive pressure.

(c) Be arranged to facilitate in-service checking for adequate venting.

(d) Have no valves in the vent between the vessel and the atmosphere.

NOTE: Inadequate venting has been the main cause of some serious explosions.

To avoid high thermal stress and fatigue for cyclically operated vessels, on/off (100% to 0%) energy control is not permitted for heat inputs greater than 1 MW.

TABLE 3.32.6

MINIMUM VENT SIZE FOR WATER HEATERS

Heat input kW	Internal diameter of vent (see Note) mm
<60	25
≥60 <150	32
≥150 <300	38
≥300 <600	50
≥600	Vent area (mm^2) equal to 3.5 times heat input (kW)

NOTE: Vents with equivalent cross sectional area may be used.

3.33 VESSELS WITH INCREASED DESIGN STRENGTH AT LOW TEMPERATURE

Design tensile strengths greater than that in Table B1 may be used for vessels operating at temperatures below 0°C, provided that:

(a) the vessels meet the requirements given in the alternative rules of ASME VIII-1-ULT for such vessels (see also Clauses 3.2.2 and 3.3.3 for further requirements for design strength at low-temperature service); or

(b) the design strength is based on reliable conservative material properties and Appendix A and is fully justified at design verification, and the parts will be stressed to these increased design strengths only when at low temperature, and are able to pass the hydro test (e.g. part of shells and supports at cryogenic temperature when supporting the associated liquid head).

3.34 PLATE HEAT EXCHANGERS

Unless otherwise agreed by the parties concerned, plate heat exchangers shall be designed as follows:

(a) End plates in accordance with Clause 3.15.

(b) Bolting in accordance with Clause 3.21.5 (for bolted types).

(c) Brazed joints in accordance with Clause 3.5 (for brazed types).

SECTION 4 MANUFACTURE

4.1 GENERAL

4.1.1 Requirements

All pressure vessels and vessel parts of welded construction shall be constructed in accordance with this Section.

4.1.2 Manufacture and workmanship

The manufacturer shall—

(a) fabricate the vessels in compliance with AS 4458;

(b) conduct or have conducted all tests and inspections of materials, processes, welding personnel and procedures during the various stages of manufacture of a vessel; and

(c) ensure that tests and inspection are witnessed and are acceptable as required by this Standard.

For guidance on purchaser's requirements see Appendix E.

4.1.3 Competence of manufacturer

The purchaser may require the manufacturer to demonstrate the adequacy of plant and personnel prior to acceptance of the manufacturer for work on vessels within the scope of this Standard.

4.1.4 Material identification and marking

Material identification and marking shall be in accordance with Clause 2.4 and AS 4458.

4.2 WELDED CONSTRUCTION

4.2.1 General welding requirements

Vessels and associated pressure parts fabricated by welding shall comply with the following requirements:

(a) All details of design and fabrication shall conform to the requirements of this Standard and AS 4458.

(b) All materials shall meet the requirements of Section 2.

(c) Welding shall be carried out by qualified personnel (Clause 4.2.2).

(d) Welding shall be carried out in conformity with the qualified welding procedure (see AS/NZS 3992).

(e) Fabrication shall be in conformity with Clause 4.1.

4.2.2 Welding personnel

4.2.2.1 *Competence of welding supervisors*

All welding shall be carried out under the supervision of a person who has had suitable training or experience in the form of fabrication and the process of welding used on the vessel, except where otherwise agreed. Such supervisor shall hold a welding supervisor's certificate in accordance with AS 1796, or have other acceptable qualifications or experience.

4.2.2.2 *Competence of welders*

Each welder engaged in the welding of vessels or vessel parts shall—

(a) hold an appropriate welder's certificate in accordance with AS 1796; or

(b) have had training or experience in the particular welding process to be used.

In addition, each welder shall meet the specific welder qualification requirements in AS/NZS 3992.

4.3 CLAD AND LINED CONSTRUCTION

Vessels or parts of vessels constructed of integrally clad plate (i.e. plate having a corrosion-resistant material integrally bonded to a base of less resistant material) and those having applied linings (i.e. a corrosion-resistant lining intermittently attached to a base of less resistant material) shall comply with the appropriate requirements of this Standard and AS 4458.

4.4 RIVETED CONSTRUCTION

NOTE: Riveted construction is rarely used for new vessels. For guidance, refer to previous editions of this Standard or AS CB1, Part 1: *Boilers other than water tube boilers and locomotive boilers for railway purposes* (superseded).

4.5 BRAZED CONSTRUCTION

4.5.1 General

Vessels and associated pressure parts fabricated by brazing shall comply with the following requirements:

(a) All details of design and fabrication shall conform to the requirements of this Standard and AS 4458.

(b) All materials shall meet the requirements of Section 2.

(c) Brazing shall be carried out by qualified personnel (see Clause 4.5.2).

(d) Brazing shall be carried out in conformity with the qualified brazing procedure (see AS/NZS 3992).

(e) Fabrication shall be in conformity with Clause 4.1.

4.5.2 Brazing personnel

4.5.2.1 *Supervisors, brazers and brazing operators*

Brazing shall be carried out under adequate supervision and by brazers and brazing operators qualified in accordance with AS/NZS 3992. Supervisors, brazers and brazing operators shall satisfy the requirements of AS/NZS 3992.

NOTE: Brazing operators are assigned to carry out brazing by automatic means or by furnace, induction, resistance, or dip brazing.

4.5.2.2 *Identification*

Each brazer and brazing operator shall be assigned an identifying number, letter or symbol by the manufacturer which shall be used to identify the work of that brazer or brazing operator.

4.5.2.3 *Record*

The manufacturer shall maintain a sufficient record of the brazers and brazing operators employed by him showing the date and results of tests and the identifying mark assigned to each. These records shall be certified by the manufacturer and shall be accessible to the Inspectors.

4.6 FORGED CONSTRUCTION

Each vessel or other pressure part constructed by forging shall comply with the relevant requirements of the forging material specification, this Standard and the requirements of AS 4458.

4.7 CAST CONSTRUCTION

Each vessel or other pressure part constructed by casting shall comply with the relevant requirement of the casting material specification, this Standard and the requirements of AS 4458.

SECTION 5 TESTING AND QUALIFICATION

5.1 GENERAL

5.1.1 Scope of section

This Section 5 specifies requirements for the qualification of welding and brazing procedures and personnel, for production testing, and for non-destructive examination and pressure tests.

5.1.2 Responsibilities and facilities for testing and inspection

The manufacturer shall be responsible for—

(a) conducting or having conducted all qualifications and tests specified in this Section;

(b) providing the labour and appliances necessary for such tests and inspection as required;

(c) such additional checks as may be agreed between the parties concerned; and

(d) giving reasonable notice to the inspection body and purchaser, as agreed between them, or when the pressure vessel will reach a stage at which inspection is required.

5.2 WELDING AND BRAZING QUALIFICATION AND PRODUCTION TEST PLATES

5.2.1 Welding and brazing procedure

Each welding and brazing procedure and each welder and brazer shall be qualified in accordance with AS/NZS 3992.

5.2.2 Welded production test plates

5.2.2.1 *General*

Welded production test plates representative of the completed vessels shall be prepared and tested to check the quality of welds in Class 1, 2A, 1H, 2H, 1S and 2S vessels except for welds listed below:

(a) Welds which comply with a prequalified welding procedure of AS/NZS 3992 where the maximum product thickness is 20 mm and TR is warmer than 0°C.

(b) Welds in group K materials with a maximum product thickness of 10 mm.

The relaxations in Items (a) and (b) do not apply to Class 2B vessels.

All conditions for the welding of production test plates shall be similar to the production welding of the vessel.

5.2.2.2 *Number of test plates for single vessels*

For each vessel as required by Clause 5.2.2.1, one production test plate welded as a continuation of a longitudinal joint shall be provided to represent each type of longitudinal weld within the limits of the essential variables of the welding procedure. This plate shall also represent circumferential joints in the same shell or ends, provided that the welding procedure is within the limits of the essential variables of the qualified welding procedure. Where one test plate represents more than one welded joint, the welding of such joints shall be carried out in a reasonably continuous operation.

An additional test plate shall be provided to represent welding where—

(a) another welding procedure outside the limits of essential variables of the first production test plate weld is used for longitudinal or circumferential type joints (Clause 3.5.1) in the main shell or ends;

(b) the length of weld represented, is evaluated for longitudinal joints only (unless a different weld procedure is used for the circumferential joint) and exceeds 200 m for automatic welding or 100 m for manual or semi-automatic welding for ferrous metals, and 30 m and 22 m respectively for non-ferrous metals; or

(c) the welding is not done in a reasonably continuous operation using the same welding procedure.

5.2.2.3 *Number of test plates for multiple vessels*

Where a number of vessels are welded in succession, one test plate may present each 200 m or fraction of automatically welded joints or each 100 m or fraction of manually or semi-automatically welded joints in ferrous metals, provided that—

(a) such test plate represents the welds within the limits of the essential variables of the welding procedure (see AS/NZS 3992); and

(b) all the welding represented by the test plate is done in a reasonably continuous operation using the same welding procedure.

5.2.2.4 *Welding and testing of test plates*

The test plates shall be welded and tested in accordance with AS/NZS 3992 and shall meet the requirements of that Standard.

5.3 NON-DESTRUCTIVE EXAMINATION

All components and welds shall be subject to non-destructive examination (NDE) as specified in AS 4037 to ensure compliance of material and manufacture with the appropriate requirements specified in that Standard. Table 1.6 describes the basic NDE requirements for each Class. NDE methods include—

(a) visual examination;

(b) radiographic examination;

(c) ultrasonic examination;

(d) magnetic particle examination;

(e) dye penetrant examination;

(f) eddy current examination; and

(g) acoustic emission examination.

5.4 Not allocated

5.5 Not allocated

5.6 Not allocated

5.7 Not allocated

5.8 Not allocated

5.9 Not allocated

5.10 HYDROSTATIC TESTS

5.10.1 General

Each vessel, after final welding and heat treatment, shall pass a hydrostatic test as prescribed in this Clause (5.10) except where the vessel is tested in accordance with Clause 5.11 (pneumatic test), Clause 5.12 (proof hydrostatic test), or Paragraph L5.3 (Class 1S and 2S vessels).

5.10.2 Test pressure

5.10.2.1 *Single-wall vessels designed for internal pressure*

For each single-wall vessel designed for internal pressure, except Class 1S and 2S vessels, and vessels provided for in Clauses 5.10.2.2, 5.10.2.4 and 5.10.2.5, the hydrostatic test pressure P_h shall be determined by the following equations:

For Class 1, 2A, 2B and 3 vessels—

$$P_h \geq 1.43P \times \frac{f_h}{f} \qquad \text{. . . 5.10.2(a)}$$

For Class 1H and 2H vessels, the greater of—

$$P_h \geq 1.43P \qquad \text{. . . 5.10.2(b)}$$

$$P_h \geq 1.25P \times \frac{f_h}{f} \qquad \text{. . . 5.10.2(c)}$$

where

P_h = hydrostatic test pressure, in megapascals

P = design pressure of the vessel, in megapascals

f_h/f = lowest ratio of—

$$\frac{\text{design strength at test temperature, MPa}}{\text{design strength at design temperature, MPa}}$$

Values of f_h and f are to be taken from Table B1 for the materials of which the main pressure-retaining parts of the vessel are constructed. For temperatures in the creep range, the design strength based on time independent properties R_{eT} and R_{mT} may be used.

The test pressure shall include any static head acting during the test on the part under consideration.

The hydrostatic test pressure may be increased by use of an additional thickness ratio, $t_n/(t_n - c)$ as in the example shown below—

$$P_h \geq 1.43P \times \frac{f_h}{f} \times \frac{t_n}{(t_n - c)} \qquad \text{. . . 5.10.2(d)}$$

where

t_n = nominal thickness, in millimetres

c = corrosion allowance in millimetres

NOTES:

1 The use of a thickness ratio to increase hydrostatic test pressure can provide increased options for the future scope of use of a vessel, or provide opportunities where engineering assessments are used to vary or determine inspection periods.

2 Where an increased test pressure is considered, the suitability of flange pressure ratings should be reassessed in order to avoid unacceptable deformation during testing.

The hydrostatic test pressure shall not exceed the following:

(a) The pressure that will result in the primary general membrane stress intensity exceeding 95% × η of the design ambient yield strength of the main shell components (where η is the weld joint efficiency).

(b) 150% of the ambient pressure rating of standard flanges listed in Clause 3.21.1(c).

NOTE: The hydrostatic test pressure can cause deformation in some cases, for example where flange ratings do not account for the hydrostatic test pressure, or where flat ends use a material whose design strength is based on yield strength and is 'strength designed' to the primary bending stress limit of 1.5*f*. Where such deformation is not acceptable (e.g. for serviceability) then a more conservative design, based on a reduced limit to primary bending stress, should be considered.

5.10.2.2 *Single-wall vessels designed for external pressure*

Single-wall vessels designed for full or partial vacuum shall be subjected to an internal hydrostatic test of at least 1.5 times the difference between atmospheric pressure (absolute) and the minimum design internal pressure (absolute). An internal vacuum test may be substituted for the hydrostatic test. The use of an internal vacuum test is the subject of agreement between the parties concerned.

NOTE: Where an internal vacuum test is used, increased non-destructive testing and leak testing of welds should be considered.

5.10.2.3 *Multiple-chamber vessels (including jacketed types)*

Vessels consisting of more than one pressure chamber shall have each chamber hydrostatically tested as follows:

(a) For chambers designed to operate independently, each chamber shall be tested at the test pressure for internal pressure or vacuum as appropriate (see Clauses 5.10.2.1 and 5.10.2.2), without pressure in the adjacent chamber.

(b) For jacketed vessels where the inner vessel is designed to operate at atmospheric pressure or vacuum condition only, the test pressure shall be determined by the equations in Clause 5.10.2.1 except that the design pressure, *P*, shall be the maximum differential pressure between inner and outer vessels, and shall be applied only to the jacket space.

(c) For jacketed vessels where the outer vessel is designed to operate at atmospheric pressure or under vacuum conditions only, the test pressure shall be determined by the equations in Clause 5.10.2.1 except that the design pressure, *P*, shall be the maximum differential pressure between the inner and outer vessels, and shall be applied only to the inner vessel. Where the outer vessel or jacket is designed to operate under vacuum conditions, it shall be tested in accordance with Clause 5.10.2.2 or pass a sensitive leak test.

(d) For chambers having common elements designed for the maximum differential pressure that can possibly occur during start-up, operation, and shut down, and the differential pressure is less than the higher pressure in the adjacent chamber, the common element shall be tested at the test pressure determined by Equation 5.10.2 except that the design pressure *P*, shall be the maximum differential pressure acting internally, where applicable. Following this test (and inspection) adjacent compartments shall be tested simultaneously at the test pressure required for internal pressure taking care to limit the differential pressure across the common elements.

5.10.2.4 *Cast iron and SG iron vessels*

The test pressure for vessels constructed of cast iron or spheroidal graphite iron shall be—

(a) 2.0 times the design pressure for design pressures exceeding 210 kPa; or

(b) 2.5 times the design pressure, but not exceeding 420 kPa, for design pressures not exceeding 210 kPa.

For vacuum and multi-chamber vessels the test pressure shall be in accordance with Clauses 5.10.2.2 and 5.10.2.3.

5.10.2.5 *Coated vessels*

Galvanized, tinned, painted, enamelled, rubber-lined or similar coated vessels shall be subjected, before coating, to a hydrostatic test to the requirements of Clause 5.10.2.1, 5.10.2.2 or 5.10.2.3, as appropriate. After coating, by agreement between the parties concerned, the vessel may be hydrostatically tested to a pressure adequate to prove the integrity of the coating but not exceeding the original test pressure.

5.10.2.6 *Tubular heat exchangers*

Tubular heat exchangers shall be hydrostatically tested at pressures in accordance with Clauses 5.10.2.1 and 5.10.2.2, as appropriate. The shell side and the tube side shall be tested separately in a manner that leaks at the tube joints can be detected from at least one side. Where construction permits, and the tube-side design pressure is the higher pressure, the tube bundle shall be tested outside of the shell.

5.10.2.7 *Clad vessels*

Vessels or parts of vessels constructed of integrally clad plate as specified in Clause 4.3 shall be tested in accordance with the relevant requirements of this Clause (5.10).

5.10.2.8 *Lined vessels*

Vessels or parts of vessels having applied linings as specified in Clause 4.3 shall be tested in accordance with the relevant requirements of this Clause (5.10) and the following:

(a) Prior to the hydrostatic test, an appropriate test shall be carried out to demonstrate the integrity of the liner. This test shall be—

    (i) a test which requires pressurizing of the space between the liner and the base metal to a pressure which will not cause buckling of the applied liner;

    (ii) a halide test;

    (iii) a penetrant test; or

    (iv) other methods agreed between the parties concerned.

    Care shall be taken to ensure the test fluid used in the space between the liner and the base material shall not cause deterioration of the vessels in service through either corrosion or the generation of excessive pressure.

    NOTE: The above test is required to minimize the potential for leakage into the space behind the liner and the need for liner repairs after the hydrostatic test.

(b) Following the hydrostatic test, the interior of the vessel shall be visually examined to determine if any seepage of the test fluid through the liner has occurred, and all welds shall be penetrant tested or otherwise tested by methods agreed between the parties concerned.

    NOTE: Where seepage of the test fluid into the space behind the liner occurs, the fluid might remain there until the vessel is put into service. Where the operating temperature of the vessel exceeds the boiling point of the test fluid, the vessel should be heated for a period sufficient to expel all test fluid from behind the liner without damaging the liner. Welding or brazing repairs should be carried out after this drying treatment.

(c) Where the integrity of the liner has been found to be defective, a suitable repair shall be made, followed by testing in accordance with methods and requirements agreed between the parties concerned.

5.10.3 Site retests

Where required by the Inspector, the completed vessel shall be hydrostatically retested on the site after erection and completion of all field welds at a test pressure agreed by the parties concerned but not less than 1.25 times the design pressure. The main component parts shall have been tested in accordance with the requirements of this Standard.

5.10.4 Tests after weld repairs

After repairs or modifications involving welding on hydrostatically tested vessels, the vessels shall be re-submitted to the standard hydrostatic test pressure, provided that in special cases or after repairs which do not affect the safety of the vessel, the hydrostatic test may be waived by agreement between the parties concerned.

NOTE: A hydrostatic test is normally necessary after repairs or modifications which—

(a) involve complete welding of portion of the main joints of the shell or ends;

(b) involve the re-welding of nozzle attachments;

(c) require re-heat treatment of the weld; or

(d) involve welding of pressure parts of carbon, carbon-manganese and alloy steel vessels where the minimum operating temperature is 30°C or more colder than the appropriate Material Design Minimum Temperature given in Figure 2.6.2.

Where a hydrostatic test is not carried out, the weld shall be subject to non-destructive testing and leak testing as agreed between the parties concerned.

5.10.5 Hydrostatic test procedure and requirements

The hydrostatic test procedure and requirement shall be in accordance with AS 4037.

5.10.6 Reporting of results

Test results shall be reported as specified in AS 4037.

5.10.7 Exemption from hydrostatic test

A vessel may be exempted from a hydrostatic test where all of the following conditions are fulfilled:

(a) The vessel contents are not classified as lethal.

(b) A hydrostatic test is likely to create a significant safety hazard because either the vessel, internal components or the supporting structure would be damaged or collapse.

(c) The vessel is Class 1 or Class 1H.

(d) The vessel is subject to 100% NDE in accordance with the relevant requirements of AS 4037.

(e) The vessels welds are additionally subject to dye-penetrant or magnetic particle examination.

(f) All nozzle to shell welds are additionally subject to ultrasonic examination.

(g) A helium leak test is performed according to Clause 5.14.

(h) The vessel is carefully inspected for damage and monitored for leaks when first pressurized during commissioning.

Exemption from hydrostatic testing is the subject of agreement between the parties concerned.

5.11 PNEUMATIC TESTS

5.11.1 General

Pneumatic testing should be avoided, but may be used in place of the standard hydrostatic test in special circumstances. Pneumatic tests and combined pneumatic/hydrostatic tests shall be in accordance with AS 4037.

NOTE: Alternatives to pneumatic testing should be considered. These might include a helium leak test, acoustic emission testing, or other alternatives.

5.11.2 Vessel quality

Butt-welded joints, nozzle welds and attachment welds shall be tested in accordance with requirements for Class 1 construction prior to pneumatic testing.

5.11.3 Test pressure

The test pressure shall be 1.25 times the equivalent design pressure or pressure difference, as required by Clause 5.10.

5.12 PROOF HYDROSTATIC TESTS

5.12.1 General

The design pressure of vessels or calculation pressure of vessel parts for which the strength cannot be calculated with a satisfactory assurance of accuracy, shall be established in accordance with the other requirements of this Clause (5.12).

The tests described in this Clause (5.12) may be used only for the purpose of establishing the calculation pressure of those parts of the vessel for which the thickness cannot be determined by means of the design requirements of this Standard.

5.12.2 Types of test

Proof tests may be of various types but provision is made here for only the following:

(a) *Strain gauge tests* (see Clause 5.12.4) and *brittle coating tests* (see Clause 5.12.5). These tests are used in all cases where it is necessary to measure local strains in selected positions in order to establish the acceptability of the design.

(b) *Displacement tests* (see Clause 5.12.6). These tests are used where it can be demonstrated that a number of displacement readings are sufficient to establish the acceptability of the design. Typical cases include measurement of the change in diameter of large nozzle to shell junctions, and circumferential measurement of cylindrical sections.

(c) *Bursting tests* (see Clause 5.12.7). These are applicable to all materials in vessels under internal pressure.

(d) *Photoelastic tests* Permissible test for the determination of critical stresses. Either two-dimensional or three-dimensional techniques may be used provided the model represents the structural effects of the loading.

(e) Combinations of the above tests.

Brittle coating and displacement tests are suitable only for vessels or parts under internal pressure, and with materials with a definite yield stress.

Tests of Items (a) and (b) are applicable only to materials having a ratio of specified minimum yield strength to specified minimum tensile strength of 0.625 or less.

5.12.3 Test arrangements

5.12.3.1 *Hydrostatic testing*

The general requirements for the relevant standard hydrostatic tests in Clause 5.10 shall apply.

5.12.3.2 *Prior pressure*

The vessel or vessel part for which the design or calculation pressure is to be established shall not previously have been subjected to a pressure greater than the anticipated equivalent design pressure (see Clause 5.10.2.1).

5.12.3.3 *Safety*

Serious consideration shall be given to the safety of testing personnel when conducting proof tests, particularly during any burst tests. Particular attention should be paid to the elimination of any air pocket.

5.12.3.4 *Witnessing of tests*

Testing shall be witnessed by a competent person. Test results shall be reported.

5.12.3.5 *Duplicate vessels*

When the design or calculation pressure of a vessel or vessel part has been established by a proof test, duplicate parts of the same materials, design and manufacture need not be proof tested but shall be given a hydrostatic test in accordance with Clause 5.10 or a pneumatic test in accordance with Clause 5.11. The dimensions and minimum thickness of the structure to be tested shall not vary materially from those actually used.

5.12.3.6 *Retests*

A retest shall be allowed on a duplicate vessel or vessel part if errors or irregularities are obvious in the test results.

5.12.4 Strain gauge tests

NOTE: See Clause 5.12.2(a) for application.

5.12.4.1 *Strain gauges*

Strains shall be measured by any device capable of measuring principal strains with a sensitivity of 5 percent and an accuracy of ±7 percent, and a range of three times the yield strain of the material being tested.

5.12.4.2 *Location of gauges*

Strain gauges shall be mounted on the unpressurized vessel before the test, and after any pressure cycling. The gauges shall be positioned in a manner that will enable strains to be measured in areas where the highest primary membrane and bending stresses are anticipated. Positioning of gauges internally and externally, selection of gauge types, rosettes, mirror pairs or the like shall be a matter for agreement between the parties concerned and the organization conducting the test. As a check that the measurements are being made at the most highly stressed areas, a brittle coating may be required to be suitably applied on all areas of probable high stress concentration.

NOTE: Strains should be measured such that they represent primary membrane stress and bending stresses. It is not intended that the design pressure obtained from this Clause be based on the measurement of high localized or secondary bending stresses. In this Standard, design requirements for details have been written to hold such stresses at a safe level consistent with experience.

The vessel may be cycled to 50% of the equivalent design pressure several times as a means of relieving most of the initial residual stress distribution.

Gauges shall be applied to the vessel before the test proper is begun and while the vessel is not pressurized.

The gauges shall be attached in a manner that will ensure accurate measurement.

5.12.4.3 *Application of pressure*

The hydrostatic pressure in the vessel or vessel part shall be increased gradually and steadily until approximately 50% of the equivalent design pressure is reached. Thereafter, the test pressure shall be increased gradually and steadily, pausing at increments of approximately 10% or less of the equivalent design pressure, until the pressure required by Clause 5.12.4.6 is reached.

The pressure shall be held stationary at the end of each increment for a sufficient time to allow the observations required by Clause 5.12.4.4 to be made.

5.12.4.4 *Strain and pressure readings*

During each pause in accordance with Clause 5.12.4.3, readings of pressure and strain at each strain gauge shall be taken and recorded. Where any such reading indicate a departure from the line of proportionality between pressure and strain, the vessel shall be completely depressurized and measurement taken of any permanent deformation at the location of each strain gauge. After such readings have been taken, the pressure shall be re-applied, as often as necessary, as specified in Clause 5.12.4.3.

5.12.4.5 *Plotting of strain*

Two curves of strain against test pressure shall be plotted for each gauge as the test progresses, one showing total strain under pressure, and the other showing permanent strain when the pressure is removed.

5.12.4.6 *Maximum test pressure*

The test may be discontinued when either—

(a) the test pressure reaches the value which will, by Equation 5.12.4.7(1) or (2) justify the desired design (or calculation) pressure;

(b) the plotted points in Clause 5.12.4.5 for the most highly strained gauge reaches 0.2% permanent strain (or 0.5% total strain for copper-base alloys, or 1.0% for austenitic stainless steel); or

(c) the measured strain becomes equivalent to 1.25 times the expected permissible design strain in the vessel component under investigation (the permissible design strain is deduced from the appropriate stress category permitted in the vessel component under investigation).

5.12.4.7 *Design pressure*

The design (or calculation) pressure to be assigned to the vessel (or part) shall not exceed the value determined by the following equation:

$$P = 0.5P_{\mathrm{h}}\left(\frac{T-c}{T}\right)\left(\frac{f}{f_{\mathrm{h}}}\right)\left(\frac{R_{\mathrm{e}}}{Y_{\mathrm{a}}}\right) \quad \ldots 5.12.4.7(1)$$

where

P = design pressure of vessel (or calculation pressure of part), in megapascals

P_{h} = hydrostatic test pressure at which test was stopped in accordance with Clause 5.12.4.6, in megapascals

T = nominal thickness of material over the area represented by the strain measurement of the most highly strained gauge, in millimetres

c = allowance for corrosion, erosion and abrasion, in millimetres

f = design strength at design temperature (see Table B1), in megapascals

f_h = design strength at test temperature (see Table B1), in megapascals

R_e = specified minimum yield strength, in megapascals

Y_a = actual average yield strength obtained in accordance with Clause 5.12.4.8, in megapascals.

As an alternative to the above and to eliminate the necessity of determining the actual yield strength of the material under test, the following equation may be used:

$$P = 0.4P_h\left(\frac{T-c}{T}\right)\left(\frac{f}{f_h}\right) \qquad \text{. . . 5.12.4.7(2)}$$

5.12.4.8 *Determination of actual yield strength*

The yield strength shall be determined in accordance with AS 1391. Materials sensitive to the rate of straining during testing shall be tested at a straining rate range (A, B or C) appropriate to the rate of straining during hydrostatic testing. The strength shall be the average of four specimens cut from the part tested after the test is completed.

The specimens shall be cut from a location where the stress during the test has not exceeded the yield strength, and shall be representative of the material where maximum stress occurs.

Where excess stock from the same piece of wrought material is available and has been given the same heat treatment as the pressure part, the test specimens may be cut from this excess stock.

Test specimens shall not be removed by thermal cutting or other method involving heat sufficient to affect the mechanical properties of the specimen.

5.12.4.9 *Interpretation of results*

In the evaluation of stress from the strain gauge data, the calculations shall be performed under the assumption the material is elastic. The elastic constant used in the evaluation of experimental data shall be that applicable to the material at the test temperature.

5.12.4.10 *Use of models for strain measurement*

Strain gauge data may be obtained from the actual vessel or from a vessel model. The material of the model need not be the same as the material of the vessel, but the material of the model shall have an elastic modulus that is the same as the material of the vessel. The requirements of dimensional similitude shall be met as closely as possible.

5.12.5 Brittle coating tests

NOTE: Brittle coating tests are rarely used for new vessels. For guidance refer to previous editions of this Standard or AS CB1, *SAA Boiler Code* (Superseded).

5.12.6 Displacement tests

5.12.6.1 *Displacement measurement*

Displacement measurement shall be made of those movements of the vessel which would arise from yielding at the location under investigation as defined in Clause 5.12.4.2. The measuring device shall be demonstrated by the manufacturer to be reliable, and accurate under all anticipated conditions of pressure, temperature variations, and vessel movement. It shall also be sufficiently sensitive and accurate to detect permanent deformation of 0.02% in the area under consideration.

5.12.6.2 *Application of pressure*

The pressure shall be applied as in Clause 5.12.4.3. However, before commencing the test proper, the vessel may be cycled to 50% of the anticipated design pressure several times as a means of relieving most of the initial residual stress distribution.

5.12.6.3 *Displacement and pressure readings*

During each pause in accordance with Clause 5.12.4.3, readings of the displacement and hydrostatic test pressure shall be taken and recorded. Where any such reading indicates a departure from the line of proportionality between pressure and displacement, the vessel shall be completely depressurized and measurement taken of any permanent deformation at the location of each displacement. Care shall be taken to ensure that the readings represent only displacement of the parts on which measurements are being made and that such readings do not include slip of the measuring devices or movement of the fixed base points or of the pressure part as a whole.

5.12.6.4 *Plotting of strain*

Two curves of displacement against test pressure shall be plotted for each reference point as the test progresses, one showing the displacement under pressure and one showing the permanent displacement when the pressure is removed.

5.12.6.5 *Maximum test pressure*

The application of pressure shall be stopped when it is evident that the curve through the points representing displacement under pressure has deviated from a straight line. The pressure coincident with the proportional limit of the material shall be determined by noting the pressure at which the curve representing displacement under pressure deviates from a straight line. The pressure at the proportional limit may be checked from the curve of permanent displacement by locating the point where the permanent displacement begins to increase regularly with further increases in pressure. Permanent deformation at the beginning of the curve that results from the equalization of stresses and irregularities in the material may be disregarded.

5.12.6.6 *Design pressure*

The design (or calculation) pressure to be assigned to the vessel or part shall not exceed the value determined by Equation 5.12.5.4(1) using P_h as the hydrostatic test pressure at which the test was stopped in accordance with Clause 5.12.6.5. Using this same designation for $P_{h,}$ the alternatives provided in Equations 5.12.5.4(2) and 5.12.5.4(3) may be used.

5.12.7 Bursting tests

5.12.7.1 *General*

NOTE: See Clause 5.12.2(c) for application.

Where the design pressure or calculation pressure is to be determined by a hydrostatic burst test, a full size sample of the vessel or part under consideration shall be used. The hydrostatic pressure shall be applied gradually and steadily, and the pressure at which rupture occurs shall be determined. This test shall not be used for vessels subject to—

(a) fatigue;

(b) external loads;

(c) thermal stresses; or

(d) corrosion.

5.12.7.2 *Design pressure*

The design (or calculation) pressure to be assigned to the vessel (or part) shall not exceed the value determined by the following equation:

$$P = \frac{P_B}{A}\left(\frac{S\eta}{S_m}\right)\left(\frac{f}{f_h}\right) \quad \ldots 5.12.7.3$$

where

A = Safety factor

$= \frac{1}{0.1 + 0.023 A_5}$

≥ 3.5

A_5 = Specified minimum percentage elongation on gauge length = $5.65\sqrt{S_o}$ or $5d$

where

S_o = original cross-sectional area, in millimetres

d = original diameter, in millimetres

NOTE: Where the minimum elongation is not available, it may be approximated by half the reduction area.

P = design (or calculation) pressure, in megapascals

P_B = bursting test pressure, in megapascals

S = specified minimum tensile strength, in megapascals

η = efficiency of welded joint (see Table 3.5.1.7), or casting quality factor

S_m = maximum tensile strength of range of specification, in megapascals

f = design strength at design temperature (see Table B1), in megapascals

f_h = design strength at test temperature (see Table B1), in megapascals

5.13 LEAK TEST

5.13.1 General

Where leak tests are specified by the purchaser on the order, they shall be carried out in accordance with this Clause (5.13).

5.13.2 Test methods

Methods of testing and acceptance criteria are the subject of agreement between the purchaser and manufacturer.

NOTE: See AS/NZS 3992 for various tests.

5.13.3 Tightness of applied linings

A test for tightness of the applied lining that will be appropriate for the intended service is recommended (see Clause 5.10.2.8).

5.13.4 Preliminary leak test

Preliminary leak tests shall be carried out in accordance with AS 4037.

5.13.5 Sensitive leak test

Sensitive leak tests shall be carried out in accordance with AS 4037.

5.14 HELIUM LEAK TEST

Helium leak tests shall be performed, where specified, after all construction, welding and heat treatment of the vessel is complete, but before any surface treatment such as painting.

Evacuation of air need not be carried out before a helium leak test is performed.

Helium used for a helium leak test shall have a purity of not less than 95%.

A helium leak test may be carried out according to the following procedure:

(a) The vessel shall be fitted with a 250 kPa calibrated pressure gauge, sealed at no greater than atmospheric pressure, and connected to a pressurizing source of helium.

(b) The vessel shall be pressurized using the helium source, to a minimum pressure of 50 kPa, and a maximum pressure that is the lesser of 200 kPa and two thirds of the design pressure.

(c) The helium source shall then be isolated (for example, by a simple valve).

(d) While the helium source is isolated, the pressure in the vessel shall be monitored for a minimum of 30 minutes to observe any pressure loss.

(e) If such a pressure loss is detected, then the source of the leak shall be identified and repaired, and the vessel shall be subject to a repeat helium leak test.

(f) Following (d) and (e), and regardless of whether any welds have been repaired, while the vessel is pressurized with helium, all welds shall additionally be inspected for any leaks by either—

    (i) soap-bubble test with copious quantities of fluid; or

    (ii) a helium gas detector.

5.15 Not allocated.

5.16 Not allocated.

5.17 SPECIAL EXAMINATIONS AND TESTS

Where specified by the purchaser, special examinations and tests shall be carried out to determine the suitability of materials or procedures new to the manufacturer. (See AS 4037.)

SECTION 6 CONFORMITY ASSESSMENT

6.1 GENERAL

This Section specifies the requirements for conformity assessment of vessels constructed to this Standard (AS 1210).

Conformity assessment comprises all checks undertaken, independently of the designer and the manufacturer, to assess conformance of the completed vessel with this Standard. It includes design verification, fabrication inspection, quality system certification, vessel testing and vessel acceptance.

6.2 ASSESSMENT

Conformity assessment shall be carried out according to one of the following:

(a) All requirements of AS 3920.1, appropriate to the vessel's hazard level determined using AS 4343.

(b) Both of the following:

 (i) The design verification requirements of AS 3920.1, appropriate to the vessel's hazard level determined using AS 4343.

 (ii) Appropriate accreditation of the manufacturer and fabrication inspector according to the requirements of the ASME Boiler and Pressure Vessel Code, with the scope of certification being relevant to the vessel.

(c) Both of the following:

 (i) The design verification requirements of AS 3920.1, appropriate to the vessel's hazard level determined using AS 4343.

 (ii) Compliance with Article 10 of the *European Pressure Equipment Directive* PED 97/23/EC appropriate for the vessel category determined using Article 9 of PED 97/23/EC.

Where independent design verification, fabrication inspection or a certified quality system is not required by AS 3920.1, the manufacturer shall ensure compliance with this Standard (AS 1210).

The selection of the conformity assessment method, and selection of the parties to carry out each part of the assessment, are the subject of agreement between the parties concerned.

NOTES:

1 Regulatory requirements for conformity assessment in some Australian jurisdictions might vary from the requirements of this Standard. Where those requirements conflict with a requirement of this Standard, users need to comply with the regulatory requirements.

2 Vessel construction should not commence until agreement has been reached on the items set out in this Clause.

6.3 CERTIFICATION OF QUALITY SYSTEMS

The certification process for manufacturing quality systems shall include the following:

(a) A general assessment of system documentation.

(b) A specific assessment of the inclusion of relevant technical and personnel requirements in the quality system.

(c) Examination of at least one example of a design or vessel, to assess the example's conformance with the requirements of this Standard (e.g. design, materials, fabrication, testing, and documentation) and the requirements set out in the quality system.

Certification bodies may apply additional elements during the certification process in order to achieve assurance of compliance.

The certification body shall have suitable, relevant national recognition.

NOTE: This might involve accreditation for the certification of pressure equipment manufacturing quality systems by JAS-ANZ (or an equivalent national accreditation body), or approval or recognition by a relevant national regulatory authority.

The certifying body should normally audit the manufacturing quality system twice per year, to ensure that the quality system is applied and maintained.

All certification processes shall be carried out by competent persons.

NOTE: See Appendix P for guidance on the judgement of competence of personnel.

The manufacturer's quality system may be certified to the requirements of the ASME Certification system, of the European Pressure Equipment Directive (PED) or of AS/NZS ISO 3834, as an alternative to certification to AS/NZS ISO 9001. The use of such alternatives is the subject of agreement between the parties concerned.

6.4 EVIDENCE OF CONFORMITY ASSESSMENT

The designer, manufacturer, and the inspection body (if used) shall provide suitable documentation of the conformity assessment.

NOTE: Requirements for vessel marking and reports are set out in Section 7.

SECTION 7 MARKING AND DOCUMENTATION

7.1 MARKING

Each completed pressure vessel complying with this Standard shall be marked with the following:

(a) Manufacturer's name or identification symbol.

(b) Identification of the inspector, inspection body, or the manufacturer's quality management system, with an appropriate mark of conformity.

(c) Design pressure, in kilopascals.

(d) Hydrostatic test pressure, in kilopascals.

(e) Date of hydrostatic test, month and year, e.g. 5/2010.

(f) Design temperature, in degrees Celsius.

(g) For vessels intended for low temperature service, the minimum operating temperature in degrees Celsius and the maximum allowable pressure at that temperature, in kilopascals.

(h) The number of this Standard and the vessel class (see Clause 1.6), e.g. AS 1210—1. For mixed class construction, use the appropriate combination of class symbols, e.g. AS 1210—1/2A. The vessel class shall be replaced with the mode of construction for brazed (B), cast (C) and forged (F) vessels.

(i) The manufacturer's serial number for the vessel.

(j) Hazard level to AS 4343.

(k) Where appropriate, the vessel registered number.

(l) Where issued by the regulatory authority, the design identification number.

(m) The appropriate units for all pressure and temperature valves marked.

NOTES:

1 When a pressure vessel is expected to operate at more than one pressure and temperature condition, other values of design pressure with the coincident design temperature may be added as required.

2 For multi-chamber vessels see Clause 7.5.

3 Additional regulatory requirements for marking may also apply.

7.2 METHODS OF MARKING

Markings shall be applied in such a way that the marking will be legible, will not be obliterated in service, and will not impair the vessel's performance.

Markings shall be stamped or etched directly on the vessel or stamped, cast or etched on a nameplate that is permanently attached to the vessel, shell or permanent part by suitable means. Nameplates attached by welding shall comply with the welding requirements of this Standard, and shall be appropriate for the expected service environment (e.g. if located on the seaboard, nameplates should be fully seal welded). For vessels constructed of Group F or Group G steel less than 12 mm thick, the markings shall be on a nameplate, except as permitted in AS 4458.

For small vessels where the above methods are impractical, the method of marking may be in accordance with AS 2971 or AS/NZS 3509.

A nameplate is recommended for—

(a) carbon and carbon-manganese steels less than 6 mm thick;

(b) non-ferrous materials less than 12 mm thick;

(c) alloy and high alloy steels;

(d) quenched and tempered alloy steels; and

(e) ferritic steels intended for use at low temperature service.

7.3 LOCATION OF MARKING

All required markings shall be located in a conspicuous place on the vessel, preferably near a manhole, other inspection opening, or other part which is accessible after installation. The marking shall be left bare, or where the vessel must be fully insulated, the insulation over the marking shall be identified and readily removable.

Markings applied directly to the vessel shall be in a position not subject to high stresses, e.g. on the rim of a flange, or thickened portion of forged ends remote from the corner radius.

7.4 SIZE AND TYPE OF MARKING

Letters and figures shall be at least 6 mm high when marked directly on the vessel, and at least 3 mm high when marked on nameplates.

The stamps for direct marking of the vessel should have letters and figures with radiused edges (i.e. 'low stress' stamps) to minimize the stress-raising effects of the stamped mark.

7.5 MULTI-CHAMBER VESSELS

Special combination units consisting of more than one independent pressure chamber or vessel shall have each pressure chamber separately marked as required for a single vessel, except where otherwise agreed.

7.6 WITNESSING OF MARKING

The marking of the vessel shall be verified by the Inspector after the hydrostatic test (or other acceptable test as per Clause 5.10.1) and other inspections have been completed to the satisfaction of the inspection body.

7.7 DOCUMENTATION

The manufacturer shall supply the purchaser with the following:

(a) A manufacturer's data report (MDR) in accordance with AS 4458 Appendix C.

(b) Additional special information required for the safe use of the vessel. Table 7.7 provides some typical types of information.

(c) Other information specified by the purchaser on the order.

NOTE: See Appendix F for information to be supplied by the manufacturer.

The manufacturer shall retain the MDR, and supporting documentation, for at least 7 years. Where a manufacturing business is sold, these records should be suitably transferred at the sale.

TABLE 7.7

GUIDELINES FOR ADDITIONAL DOCUMENTATION

Type of documentation	Hazard level (To AS 4343)*				
	A	B	C	D	E
MDR and conformance certificate	Y	Y	Y	Y	Y
Instructions and limitations, such as purchase specifications, operational assumptions, specific design requirements such as nozzle loads, environmental loads, foundation loads	Y	Y	Y	Y	Y
Design verification certificate	Y	Y	Y	Y	—
Calculations and design report	Y	Y	Y	—	—
Material certificates	Y	Y	Y	—	—
NDE reports	Y	Y	Y	Y	—
Welding procedures and qualifications	Y	Y	Y	Y	—
Welder identification and qualifications	Y	Y	Y	—	—
General assembly drawings	Y	Y	Y	Y	—
Detail drawings	Y	Y	Y	—	—
Lifting and transport drawings	Y	Y	Y	—	—

* Y = Documentation is recommended.

SECTION 8 PROTECTIVE DEVICES AND SYSTEMS

8.1 GENERAL REQUIREMENTS

8.1.1 General

Each pressure vessel shall be provided with protective devices and other fittings in accordance with the requirements of this Section.

The number, size, type, location and performance of protective devices and other fittings required by this Standard and for the safe operation of the vessel is the subject of agreement between the parties concerned.

NOTES:

1 It is expected that the necessary devices and other fittings are supplied and fitted prior to placing the vessel into service.

2 The presence of electronic and electrical control systems, and the possibility of different operational conditions, transient conditions and failures, can influence the protective devices or systems needed to ensure reliable performance of the vessel, particularly in complex installations or applications. Such information should be discussed between the parties concerned.

8.1.2 Design, manufacture and connection for protective devices and fittings

Protective devices, and other fittings shall be of material, design and manufacture to permit the devices to perform their required function under the expected conditions of service. They shall be acceptable to the purchaser and, where required, to the inspection body and where appropriate shall comply with AS 1271 or equivalent.

All connections to the vessel shall comply with the requirements of Clause 3.19.

8.2 VESSELS REQUIRING PRESSURE-RELIEF DEVICES

8.2.1 Pressure relief—General requirements

Each pressure vessel shall be protected with one or more pressure-relief devices except as provided in Clauses 8.2.5, 8.2.6 or 8.2.7.

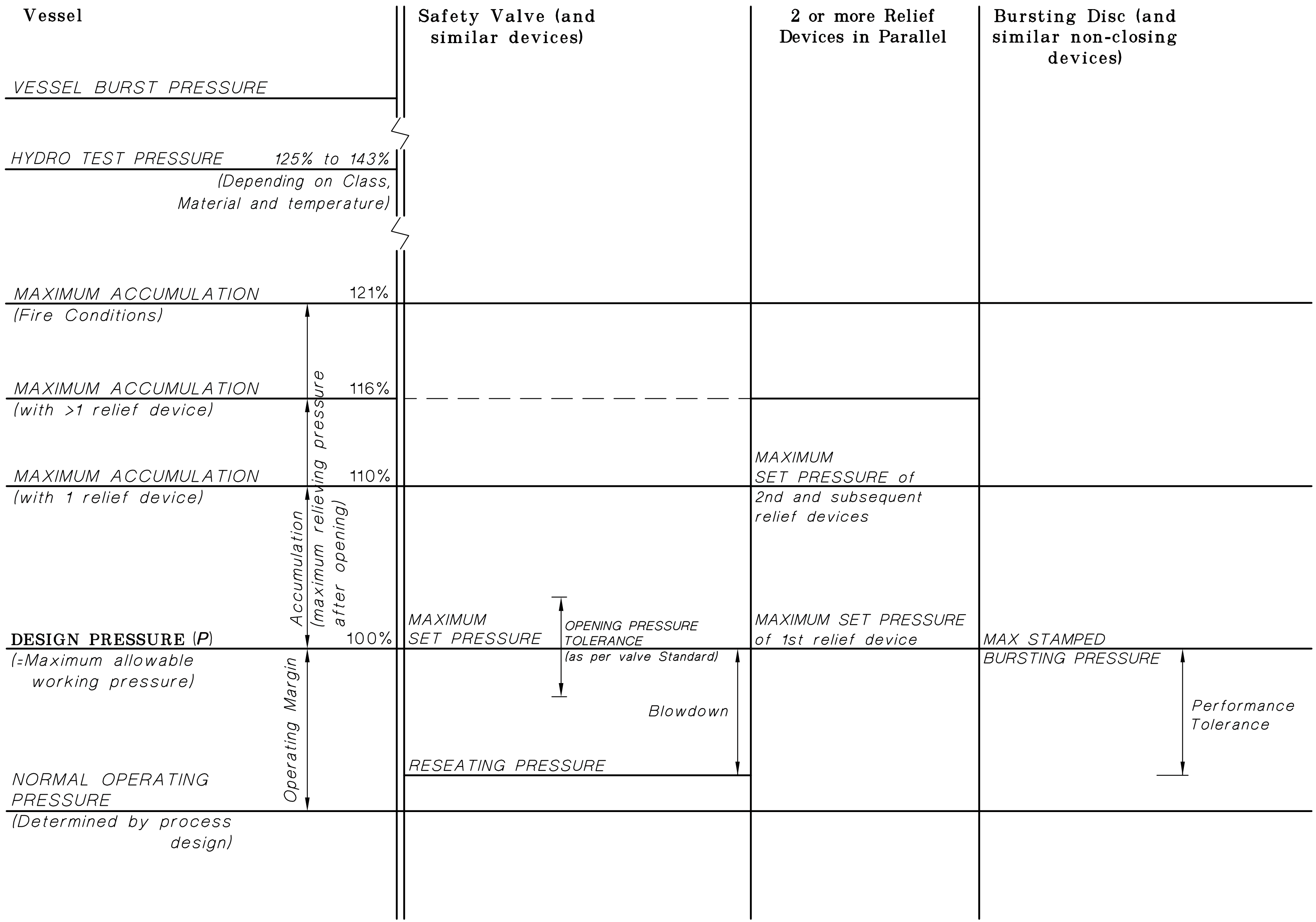
Each chamber or compartment of a multi-chamber vessel (including jackets) shall be treated as a separate vessel and shall be suitably connected to a pressure-relief device unless the compartments are interconnected in accordance with Clause 8.2.4.

Each pressure vessel shall be protected by a pressure-relieving device that shall prevent the pressure from rising to more than 110% of the design pressure of the vessel except as follows (see Clause 8.7 for requirements for setting of pressure relief devices):

(a) Where multiple pressure-relief devices are provided and set in accordance with Clause 8.7.1, they shall prevent the pressure within the vessel rising to more than 116% of the design pressure, provided that the lower set pressure-relief device(s) is capable of relieving any surge condition anticipated during normal operation.

(b) Where excess pressure is caused by exposure to fire or other unexpected source of heat, the pressure-relief device(s) shall comply with Clause 8.2.2.

(c) Where the relevant application code specifies otherwise (e.g. AS/NZS 1596 or AS/NZS 2022).



NOTE: This Figure is for information only. Requirements are set out in the relevant clauses of this Standard.

FIGURE 8.2 GUIDE TO DESIGN AND RELIEVING PRESSURE AT DESIGN TEMPERATURE

8.2.2 Pressure relief for fire conditions

Where an additional hazard can be created by exposure of a vessel to fire or other similar unexpected source of heat (e.g. vessels used to store flammable liquefied gases), the pressure-relief devices shall be capable of preventing the pressure from rising to more than 121% of the design pressure of the vessel.

Requirements associated with fire conditions are set out in Clauses 8.6.2, 8.7.3, 8.11 and 8.12.

The same pressure-relief device may be considered to meet the requirements of both Clauses 8.2.1 and 8.2.2 provided it meets the individual requirements of each clause.

8.2.3 Liquid full vessels

Vessels that are to operate completely filled with liquid shall be equipped with a liquid-relief valve(s) unless otherwise protected against over-pressure. (See Clause 8.3(a)(ii).)

8.2.4 Interconnected vessels and chambers

Vessels or chambers of vessels, which are connected together in a system by piping of adequate capacity, may be considered as one unit in determining the number and capacity of pressure-relief devices provided that no valve is fitted that could isolate any vessel from the relief devices unless that vessel is simultaneously opened to atmosphere.

8.2.5 Systems of limited or reduced pressure

Where the source of pressure is external to the vessel and is under such positive control that the pressure in the vessel cannot exceed the design pressure at the operating temperature, the requirements of Clause 8.2.1 above need not apply, but suitable provision shall be made to comply with Clause 8.2.2.

Pressure-reducing valves and similar mechanical or electrical pressure control instruments, except for pilot-operated relief valves as permitted in Clause 8.4.4, are not considered as sufficiently positive in action to prevent excess pressures from being developed.

8.2.6 Lethal fluids and other special fluids

Under special conditions of service and by agreement between the parties concerned, vessels containing lethal fluids or other special fluids may be exempt from the requirement of this Section.

8.2.7 Safety instrumented systems (SIS)

8.2.7.1 *General*

Where it is not practical to use conventional pressure relief valves and fittings, or where such equipment is demonstrated to be unable to provide adequate protection against over-pressure, then a safety instrumented system (SIS) may be used to control and manage overpressure. Such a system shall comply with AS 61508 and AS IEC 61511.

NOTE: A SIS is a designated system that both—

(A) implements the required safety functions necessary to achieve or maintain a safe state for the equipment under control; and

(B) is intended to achieve, on its own or with other electrical/electronic programmable electronic safety-related systems, other technology safety-related systems or external risk reduction facilities, the necessary safety integrity for the required safety functions.

8.2.7.2 *Requirements for safety instrumented systems*

An SIS to be used for protection against overpressure shall be developed using the safety life cycle (SLC) approach as follows (refer AS 61508):

(a) An analysis of the situation including the identification of hazards relating to the process and the equipment, thereby obtaining a thorough understanding of the risks involved.

(b) Documentation of target safety integrity levels (SIL) and safety requirements for the equipment concerned.

(c) Development of a documented safety system design, using appropriate design methodology and hardware/software subsystems as applicable.

(d) Evaluation against the required safety integrity level (SIL) and reliability specifications, then modified as necessary.

(e) Operation and maintenance according to accepted procedures.

(f) Documentation of results to ensure that performance standards are maintained throughout the systems life.

8.3 TYPES OF PRESSURE-RELIEF DEVICES

Pressure-relief devices are devices designed to relieve excess pressure, and for the purpose of this Standard are of the following types:

(a) *Pressure-relief valve* A safety valve or relief valve as defined in Item (i) or Item (ii).

  (i) *Safety valve*—a valve which automatically discharges fluid to atmosphere so as to prevent a predetermined pressure being exceeded. It is normally used for compressible fluids which require quick over-pressure relief. It is activated by the static pressure upstream of the valve.

  NOTE: These valves may also be referred to as safety relief valves when they are suitable for use as a safety valve or a relief valve depending on application.

  (ii) *Relief valve*—a valve which automatically discharges fluid to atmosphere or a reduced pressure system so as to prevent a predetermined pressure being exceeded. It is used primarily for non-compressible fluids (i.e. liquids). It is activated by the static pressure upstream of the valve.

NOTE: The valves in Items (i) and (ii) are designed to reclose after normal conditions have been restored.

(b) *Bursting disc and other non-reclosing pressure relief device* A bursting disk type pressure relief device has an operating part in the form of a disc or diaphragm normally of metal, which initially blocks a discharge opening in the vessel but bursts at a predetermined pressure to discharge fluid. It does not re-close automatically.

Other non-reclosing pressure relief devices include breaking-pin devices, buckling-pin devices, spring loaded non-closing pressure relief valves and for vacuum jackets, pressure relief plugs or plates, which have similar function to bursting disc.

(c) *Vent system* Where a vessel is open to atmosphere through a vent pipe (with or without a liquid trap) the vent pipe may be regarded as a pressure (or vacuum) relief device, provided that the vent system meets the requirements of Clause 8.2.1, is connected as directly as possible to atmosphere, is used for this purpose only, and cannot be closed or blocked by ice formation or collection of deposits.

(See Clauses 8.11, 8.12 and 8.13 for other protective devices that may limit pressure.)

8.4 PRESSURE-RELIEF VALVES

8.4.1 Application

In general, pressure-relief valves are preferred for protection of vessels against excessive pressure but bursting discs or other non-reclosing pressure relief devices may be used as agreed. If the fluid is compressible, a pressure-relief valve and bursting disc may be arranged in series, and such arrangement may be preferable as indicated in Clause 8.5.1.

8.4.2 Design, manufacture testing and marking

The design, manufacture, testing and marking of pressure-relief valves shall comply with AS 1271.

8.4.3 Type and minimum bore

Pressure-relief valves shall be of the spring-loaded type, however deadweight-type valves may be used for static vessels by special agreement between the parties concerned. Lever and weight types shall not be used.

The minimum bore (see Note) for any relief valve used on a vessel shall be as follows:

(a) For steam where some corrosion or deposit may cause blockage or sticking.....10 mm;

(b) For a gas or vapour which may cause sticking of the valve disc ....10 mm;

(c) For other gas or vapour... 6 mm;

(d) For a liquid which may cause sticking of the valve disc ....20 mm;

(e) For other liquid..12 mm.

NOTE: The bore is the diameter at the minimum cross-sectional flow area (but not the curtain area) between the inlet and the seat with no reduction for any obstruction. It is sometimes referred to as the orifice diameter or by a letter designation representing its equivalent effective area.

To limit loss of fluid with any short term over pressure and to provide a back-up in the event of valve sticking or blockage, it is recommended that where one valve is fitted, its bore (see above Note) not exceed 75 mm nominal, unless otherwise agreed. If a discharge capacity greater than that provided by one 75 mm nominal, or other accepted valve is necessary, then two or more valves should be fitted. Where more than one valve is fitted, one or more such valves may exceed 75 mm nominal, provided that the discharge capacity of the smallest valve is not less than the smaller of 50% of the discharge capacity of the largest valve or 25% of the required total discharge capacity.

8.4.4 Pilot operation

Pilot-valve control or other indirect operation of safety valves is not permitted as part of the required pressure-relief system and contributing to the required relieving capacity unless—

(a) the design is acceptable to the purchaser and the inspection body;

(b) the fluid to be relieved is clean vapour or pilot purges are used to ensure that the pilot valve is kept clear; and

(c) the design is such that the main valve will open automatically at not over the set pressure and will discharge its full rated capacity if some essential part of the pilot or auxiliary device should fail to operate, or the complete valve is designed to have fail-safe characteristics approaching those of the above type of system.

8.4.5 Easing gear

Easing gear shall be fitted to pressure-relief valves for use with steam, air and those fluids which promote sticking of the valve disc to the seat but do not create a hazard when released, (e.g. leakage of the fluid is prevented at all places other than through the discharge piping to a safe location).

Easing gear shall be such that the disc can be positively lifted off its seat when the valve is subjected to its set pressure minus 690 kPa or to 75% of its set pressure, whichever is the higher pressure.

8.4.6 Gumming and thermal effects

The design of pressure-relief valves and the choice of their materials of manufacture shall take into consideration the possible effect of differential expansion and contraction, of possible icing of external components during discharge and of gumming or deposits. Valves with plain flat discs without bottom guides shall be used when gumming or deposits are probable on the inside. The valve spring should be protected by a suitable seal where it is liable to corrosion or blockage due to products discharged.

8.4.7 Drainage

Where liquid can collect on the discharge side of the disc of a pressure-relief valve, the valve shall be equipped with a drain at the lowest point where liquid can collect.

8.4.8 Vapour tightness

For toxic or flammable fluids safety and relief valves shall meet the requirements for vapour tightness where specified by the purchaser.

8.5 BURSTING DISCS AND OTHER NON-RECLOSING PRESSURE-RELIEF DEVICES

8.5.1 Application

Bursting discs or a combination of bursting discs and other pressure-relief devices (see Clause 8.3) are recommended for the following conditions, as relevant:

(a) Where pressure rise may be so rapid as to be analogous to combustion or explosion.

(b) Where even minute leakage of the fluid cannot be tolerated during normal service, e.g. with highly toxic or valuable materials.

(c) Where service conditions may involve heavy deposits or gumming up such as could render a pressure-relief valve inoperative.

For vacuum jackets, a suitable pressure relief plate or plug may be used as a bursting disc.

Where a system is subject to pulsating pressure, reversal of pressure, corrosion, or elevated temperatures, bursting discs shall be used with caution. There should also be a substantial margin between the maximum operating pressure and the bursting pressure of bursting discs (see Clause 8.7.2).

NOTE: A register of bursting disc data is to be kept by the user for each vessel protected by a bursting disc. The register should relate the service conditions at which a vessel operates to the serial letters and numbers stamped on the disc or marked on the envelope in which the disc was contained.

8.5.2 Design, manufacture, testing and marking

The design, manufacture, testing and marking of bursting discs shall comply with AS 1358.

8.5.3 Discs located between pressure-relief valve and vessel

A bursting disc may be installed between a spring-loaded pressure-relief valve and the vessel provided that—

(a) the valve is ample in capacity to meet the requirements of Clause 8.6;

(b) the maximum pressure of the range at which the disc is designed to burst does not exceed the design pressure of the vessel;

(c) the discharge capacity of the bursting disc after rupture is not less than the capacity of the associated valve;

(d) the open area of the bursting disc after rupture is no less than the inlet area of the valve;

(e) following rupture there is no possibility of interference with the proper functioning of the valve; and

(f) the space between the bursting disc and the valve is provided with a pressure gauge, trycock, free vent, or other suitable indicator for the detection of disc rupture or leakage.

NOTE: Users are warned that a bursting disc will not burst at its design pressure or may fail in reverse bending if back-pressure builds up in the space between the disc and the relief valve, e.g. where leakage develops in the bursting disc due to corrosion or other causes.

8.5.4 Disc located on discharge side of pressure relief valve (see Note 2)

A bursting disc in series with the pressure-relief valve may be used to minimize the loss by leakage through the valve of hazardous materials, and where a bursting disc alone or a disc located on the inlet side of the safety valve is impracticable. The distance between the valve and the bursting disc shall be a practical minimum.

A bursting disc may be installed on the outlet of a spring-loaded pressure-relief valve which is opened by direct action of the pressure in the vessel provided that—

(a) the valve is of a type which will open at its set pressure regardless of back pressure (see Note 1);

(b) a valved vent is located between the valve disc and the bursting disc to permit venting of the valve to a safe location;

(c) the valve is ample in capacity to meet the requirements of Clause 8.6;

(d) the maximum pressure of the range for which the disc is designed to burst does not exceed the design pressure of the vessel (see also Item (k));

(e) the discharge capacity of the bursting disc after rupture is not less than the capacity of the associated valve, and the open area through the disc after rupture is not less than the outlet area of the valve;

(f) piping beyond the bursting disc cannot be obstructed by the bursting disc or fragments;

(g) all valve parts and joints subject to stress due to the pressure from the vessel and all fittings up to the bursting disc are designed for not less than the maximum operating pressure of the vessel;

(h) any small leakage or a larger flow through a break in the operating mechanism that may result in back-pressure accumulation within enclosed spaces of the valve housing other than between the bursting disc and the discharge side of the pressure-relief valve, so as to hinder the pressure-relief valve from opening at its set pressure, will be relieved adequately and safely to atmosphere through telltale vent openings;

(i) the content of the vessel is a clean fluid, free from gumming or clogging matter, so that deposits in the space between the valve and the bursting disc (or in any other outlet that may be provided) will not clog the outlet;

(j) the installation is acceptable to the parties concerned; and

(k) the bursting pressure at atmospheric temperature does not exceed the maximum operating pressure of the vessel at atmospheric temperature.

NOTES:

1 Users are warned that an ordinary spring-loaded pressure-relief valve will not open at its set pressure if back pressure builds up in the space between the valve and bursting disc. A specially designed pressure-relief valve is required, such as a diaphragm valve or a valve equipped with a bellows above the disc.

2 Users are warned that replacing a bursting disc on the outlet of a pressure-relief valve may be attended by some danger if done without first reducing the pressure in the vessel, particularly when hazardous contents might be discharged.

8.5.5 Other non-reclosing pressure relief of devices

These devices shall comply with similar requirements as for bursting discs and shall—

(a) be fully open at the set pressure;

(b) have a set pressure tolerance not greater than ±5%;

(c) be limited to service temperature of −30°C to 150°C for buckling pin devices;

(d) have a calculated discharge capacity based on the minimum discharge area and a discharge co-efficient not greater than 0.62 unless a higher value is proven by adequate tests; and

(e) be suitably protected from contamination or interference.

8.6 REQUIRED DISCHARGE CAPACITY OF PRESSURE-RELIEF DEVICES

8.6.1 Aggregate capacity

The aggregate capacity of the pressure-relief devices connected to a vessel (or system of vessels) shall be sufficient to allow the discharge of the maximum quantity of fluid that can be generated or supplied to the vessel (in a credible relief scenario), without giving rise to a pressure exceeding the maximum limits specified in Clause 8.2.1 and Clause 8.2.2. (See Clause 8.2.1(a) for required relieving capacity of the lowest set pressure-relieving device.)

NOTE: A safety instrumented system (SIS) developed with regard to the requirements of Clause 8.2.7 is an acceptable method of limiting the maximum quantity of fluid that can be generated or supplied to the equipment during relief scenarios.

8.6.2 Aggregate capacity for fire conditions

8.6.2.1 *General*

Relief devices required by Clause 8.2.2 as protection against fire or other unexpected sources of external heat shall have relieving capacity sufficient to prevent pressure from rising to more than 121% of the vessel design pressure. The relieving capacity shall be determined in accordance with Clauses 8.6.2.3 or 8.6.2.4. See also Clause 8.12.

8.6.2.2 *Notation*

For the purpose of this Clause (8.6.2), the notation given below applies:

A_e = external area of vessel adjacent to the maximum feasible wetted area below 7.5 m height above any potential sizeable source of flame or heat, in square metres which may be taken as:

(a) For cylindrical vessel with spherical ends—

$\pi \times$ overall length $\times$ outside diameter.

(b) For cylindrical vessel with 2:1 ellipsoidal and torispherical ends—

$\pi \times$ (overall length + 0.19 outside diameter) $\times$ outside diameter.

(c) For spherical vessel—

π (outside diameter)^2.

NOTE: The source of flame or heat usually refers to ground level but may be at any level at which a sizeable fire could be sustained.

C = constant for gas

$$= 3.948\left[k\left[\frac{2}{k+1}\right]^{(k+1)/(k-1)}\right]^{1/2}$$

NOTE: $3.948 = \dfrac{3600\,(\text{h to s}) \times 0.1\,(\text{bar to MPa})}{R^{0.5}}$

C_w = specific heat per unit volume of vessel wall, in kilojoules per cubic metre kelvin

= 2425 for aluminium

= 3550 for steel

= 3970 for nickel

= 3430 for copper

F = insulation factor

The following are recommended minimum values of F, but may require adjustment where special conditions exist:

For bare uninsulated vessel 1.0

For insulated vessels having thermal conductance at 889 K temperature difference of:

22.7 W/m^{2}K 0.3

11.4 W/m^{2}K 0.15

5.7 W/m^{2}K 0.075

For underground vessels 0

For aboveground vessels covered with earth 0.03

For bare vessels with water sprays 1.0

For transportable vacuum-insulated vessels where the outer shell will remain completely in place when subjected to 650°C 0.0132U

where

U = total thermal conductance of the container insulating material, in watts per square metre kelvin, when saturated with gaseous cargo or air at atmospheric pressure, whichever is greater. The value of U shall take into account any heat flow through nozzles and supports.

For transportable foam-insulated vessels where the foam will remain substantially in place when subjected to 650°C .... $0.1 + 0.01188U$

U = total thermal conductance of the foam insulation, in watts per square metre kelvin, assuming that the insulation has lost 25 mm of its thickness and is saturated with gaseous cargo or air at atmospheric pressure, whichever provides the greater value of thermal conductance. The value of U shall take into account any heat flow though nozzles and supports.

NOTE: This factor is based on the assumption that all insulation has been removed over 10 percent of the total vessel surface area.

f = design stress of vessel wall at design temperature (from Table B1), in megapascals

k = isentropic exponent (ratio of specific heats for constant pressure and volume) for gas, $\frac{C_p}{C_v}$

L = latent heat of vaporization of vessel contents, in joules per kilogram

M = molecular weight of stored fluid, in kilograms per kilomole

m = maximum mass of stored gas in vessel, in kilograms

m' = initial flow of gas when relief device opens at relieving conditions, in kilograms per second

m'_p = maximum gas flow from plant and compressor into the vessel, in kilograms per second

p = vessel design pressure, in megapascals

Q_a = minimum required aggregate vapour capacity of relief devices, in cubic metres per minute of air at 15°C and 101.5 kPa (abs)

R = Universal gas constant

= 8314, in joules per kilomole kelvin

T = design temperature, in kelvin

T_o = minimum operating temperature at design pressure, in kelvin

T_r = relieving temperature, in kelvin

= temperature corresponding to $1.21p + 0.1$ megapascals (absolute) on vapour saturation curve for vessels containing liquified gas or liquids

= relieving temperature of any temperature relief device for vessels containing gas, which shall not exceed the temperature corresponding to $\frac{Z}{1.21} \cdot f$ (used in the design) on the f versus T curve or on the 1/1.3 creep rupture strength (mean in 2 hours) versus T curve as agreed by the parties concerned

t = corroded vessel wall thickness, in millimetres

Y_t, Y_p = relieving constant, in (seconds)$^{-1}$

Y_t = $110\,000/C_w t T_r$

Y_p = 10 000/$C_w t T_o$

Z = compressibility of gas or vapour at relieving conditions

n = weld joint efficiency

8.6.2.3 *Fire relief of vessels containing liquefied gas or liquid*

The minimum total discharge rate of the relief devices shall be:

$$m' = \frac{7.2\times10^4 F\ A_e^{0.82}}{L} \qquad \text{. . . 8.6.2.3(1)}$$

Where drainage beneath the vessel will avoid collection of large quantities of flammable material and other site factors will limit the intensity or proximity of a fire, the relief valve capacity for static vessels determined by the above equation may be reduced by up to 40 percent.

Thus the minimum total discharge capacity of the relief devices shall be:

$$Q_a = 41.44\ \frac{m'}{C}\left[\frac{T_r Z}{M}\right]^{1/2} \qquad \text{. . . 8.6.2.3(2)}$$

8.6.2.4 *Fire relief of vessels containing gas or vapour*

For vessels fully or partially filled with gas or vapour above its boiling point under fire conditions, excessive distortion or loss of containment cannot always be prevented by a pressure relief device only. Excessive temperature may weaken the vessel wall enough to cause distortion or rupture before or during relief device operation.

The following requirements apply:

(a) *Inherent protection* Special fire protection devices are not required on vessels—

(i) where the location and contents are such that they cannot experience an accidental heat flux exceeding 10 kW/m^2;

(ii) that do not pose any other unacceptable risk due to loss of containment (e.g. does not release large quantities of toxic, flammable, or combustible fluid);

(iii) that are insulated with fireproof insulation having thermal conductance of less than 20 *W*/m^{2}K at 800°C; or

(iv) that have shown from experience or tests to suitably withstand fire.

NOTE: Experience with grass, bush, vehicle and building fires indicates that steel LP Gas vessels can withstand full engulfment in a high intensity fire for at least 15 minutes before rupture or release of contents, and so facilitate emergency action. Thus for low intensity or short duration fires, no special fire protection is necessary (see also Clause 8.12).

(b) *Over-temperature protection* Vessels requiring fire protection shall be protected by both temperature and pressure sensitive relief devices or comply with Clause 8.12.

Temperature sensitive relief devices shall be sized so that the initial flow under fire conditions is at least:

$$m' = mY_t + m'_p \qquad \text{. . . 8.6.2.4(1)}$$

Such temperature sensitive relief devices may take the form of fusible elements which melt at or below T_r, or valves actuated by temperature sensors e.g. thermocouples set at T_r or gaskets and seals which leak on exposure to fire. In any case the design of such temperature sensitive relief devices shall have the following features:

(i) Their position, number and distribution of sensors around a vessel shall provide early detection of high wall temperature to prevent excessive thermal weakening.

(ii) For temperature actuated relief valves (i.e. other than fusible elements), the components of the relief system exposed to fire shall have a minimum fire rating of 30 minutes.

(c) *Over-pressure protection* Over-pressure protection devices shall be sized so that the initial flow is at least:

$$m' = mY_{\mathrm{p}} + m'_{\mathrm{p}} \qquad \text{. . . 8.6.2.4(2)}$$

Such protection is afforded by conventional pressure relief devices.

(d) *Alternative* As an alternative to Items (b) and (c), the method given in ANSI/API RP 520 may be used.

8.6.3 Capacity for burst tube

Where a vessel is fitted with a heating coil or other element whose failure might increase the normal pressure of the fluid in the vessel, e.g. in heat exchangers, calorifiers and evaporators, and the shell design pressure is lower than the design pressure of such element, the relieving capacity of the pressure-relief device shall be adequate to limit the increase in the shell pressure in the event of such failure.

Such vessels which are liquid-filled in both the shell and tubes and which may be subject to shock loading in the event of tube failure shall be fitted with a bursting disc or similar device of size determined by Equation 8.6.3(1). Such bursting disc shall be in addition to the otherwise required pressure-relief devices.

$$A = 2a\left(\frac{P_{\mathrm{t}} - P_{\mathrm{v}}}{P_{\mathrm{v}}}\right)^{1/2} \qquad \text{. . . 8.6.3(1)}$$

where

A = minimum effective area of bursting disc, in square millimetres

a = area of bore of one tube, or of inlet pipe to tubes, or of any restricting orifice fitted in the inlet side, whichever is least, in square millimetres

P_{t} = design pressure of tubes, in megapascals

P_{v} = design pressure of vessel shell, in megapascals

For other vessels including evaporators and similar vessels, the safety valves shall have a discharge capacity sufficient to limit pressure under normal operation and shall have a minimum total effective discharge area determined by the following equation:

$$A = \frac{2a}{1.10}\left(\frac{P_{\mathrm{t}} + 0.1}{P_{\mathrm{v}} + 0.1}\right) \qquad \text{. . . 8.6.3(2)}$$

where

A = minimum total effective discharge area of safety valves, in square millimetres

Where a restricting orifice is fitted to limit the flow which the pressure-relief device is required to relieve, it shall be constructed of corrosion-resistant material and shall have the orifice parallel for at least 6 mm.

8.6.4 Capacity for calorifiers and similar vessels

The capacity of pressure-relief valves for these vessels shall be based on the manufacturer's rated output of the calorifier and shall be at least equal to the maximum steam capacity capable of being developed at 110% of the design pressure. The minimum required discharge capacity may be determined as follows:

$$\text{Capacity, kg/h} = 5.4\times10^{6}\,\frac{R}{L}\left(\frac{T_1-T_2}{T_1-T_3}\right) \qquad \text{. . . 8.6.4}$$

where

R = rated capacity, in kilowatts

L = latent heat of steam at 1.10 times set pressure, in joules per kilogram

T_1 = design steam temperature in tubes, in degrees Celsius

T_2 = saturation steam temperature at 1.10 times set pressure, in degrees Celsius

T_3 = design water temperature in shell, in degrees Celsius

For vessels using fluids other than water, or steam, the capacity of relief devices shall be based on the same principles.

8.6.5 Certified capacity of safety and relief valves

The capacity of safety and relief valves selected to meet the requirements of this Section (8.6) shall be the capacity certified in accordance with AS 1271 as adjusted to suit the particular fluid concerned, using data in AS 1271.

8.6.6 Liquid relief capacity of pressure-relief devices

The capacity of pressure-relief devices discharging liquid, selected to meet the requirements of this Section, may be the manufacturer's rated capacity determined in accordance with AS 1271 as adjusted to suit the particular fluid concerned.

8.6.7 Capacity for refrigerated or vacuum-insulated vessels

The capacity of pressure-relief devices for refrigerated, insulated, and vacuum-insulated vessels shall provide adequate venting capacity to meet requirements of Clauses 8.6.1 and 8.6.2 on the basis of failure of refrigeration systems or saturation of insulation space with vessel contents or with air at atmospheric pressure.

8.7 PRESSURE SETTING OF PRESSURE-RELIEF DEVICES

8.7.1 Pressure-relief valves

Where one or more pressure-relief valves are fitted, at least one valve shall be set to discharge at or below the design pressure of the vessel, except as permitted in Clause 8.7.3. Any additional valves fitted may be set to discharge at a pressure not exceeding 110% of the design pressure provided the aggregate valve capacity complies with Clause 8.6.1. (See Clause 3.2.1.1 concerning margin between set pressure and maximum working pressure.)

8.7.2 Bursting discs

Bursting discs fitted in place of or in series with pressure-relief valves shall have a maximum marked burst pressure such that rupture of the disc will occur at a pressure not exceeding the design pressure of the vessel at the operating temperature (see Clause 3.2.1.1).

Where a bursting disc is fitted in parallel with relief valves to protect the vessel against explosion hazard and is not required to contribute to the required aggregate relieving capacity, the disc may have a maximum bursting pressure at atmospheric temperature (i.e. specified bursting pressure plus positive tolerance) not greater than the standard hydrostatic test pressure for the vessel.

8.7.3 Pressure-relief devices for fire conditions

Pressure-relief devices permitted by Clause 8.2.2, where exposure to fire or other unexpected sources of external heat is the only possible source of overpressure, shall be set to open at a pressure not exceeding 110% of the design pressure of the vessel unless otherwise permitted by the application Standard.

NOTE: This increased opening pressure is not intended to apply to 'pop-up' valves used with liquefied gas vessels, as such valves are designed to begin to open at the design pressure, and reach full discharge capacity at 121% of design pressure.

If such a device is used to meet the requirements of both Clauses 8.2.1 and 8.2.2 it shall be set to open in accordance with Clause 8.7.1.

8.7.4 Superimposed back pressure

The pressure at which a pressure relief device is set to open shall take into account the effect of superimposed back pressure.

8.7.5 Minimum set pressure

Where vessels contain flammable or toxic materials which may cause a hazard in the event of safety devices venting, the set pressure of safety devices shall be as high as practicable and as permitted by this Section.

8.8 INSTALLATION OF PRESSURE-RELIEF DEVICES

8.8.1 Safety valves and non-reclosing devices

Safety valves, bursting discs and other non-reclosing relief devices shall be connected to the vessel in the vapour space above any contained liquid, or to piping connected to the vapour space in the vessel which is to be protected. Safety valves shall be mounted with the spindle vertical and pointing up, except that for valves not exceeding 32 mm nominal bore, other spindle positions may be used, provided that the installation complies with the valve manufacturer's recommendation. With vessels containing viscous liquids, special precautions shall be taken to place safety valves in a position where contact with such fluid will not prevent the valve from performing satisfactorily.

8.8.2 Relief valves

Relief valves for liquid service shall be connected below the normal operating liquid level.

8.8.3 Inlet connection

To ensure the inlet arrangement to the relief device does not seriously affect the device's performance, the connection between the relief device and the vessel shall be as short and straight as practicable, shall have a bore area at least equal to the area of the relief device inlet, and shall not reduce the discharge capacity of the relief device below the capacity required for the vessel. Where the relief device is not close to the vessel, allowance shall be made for the pressure loss from the vessel to the relief device orifice. For conventional safety valves that do not preclude rapid opening and closing, the (non-recoverable) inlet pressure loss shall not exceed three percent of the set pressure (based on the steady flow capacity of the valve, i.e. excluding short term transient effects).

The opening in the vessel wall shall be designed to provide direct and unobstructed flow between the vessel and its pressure-relief device. Rounding the edges of the entrance will assist in limiting the pressure drop to the device.

When two or more required pressure-relief devices are placed on one connection, the internal cross-sectional area of this connection shall be at least equal to the combined inlet areas of the relief devices connected to it and in all cases shall be sufficient to avoid restriction of the combined flow of the attached devices.

The inlet connection shall be arranged to prevent collection of foreign matter or liquid at the inlet to the relief device, and should be located where turbulence is not excessive.

No connection other than such as will cause no flow (e.g. pressure gauge) shall be made between the vessel and its relief device.

8.8.4 Stop valves between pressure-relief device and vessel

Unless the installation complies with the requirements of either Item (a) or (b) hereof, no means of isolation shall be provided between the vessel and any pressure relief device.

(a) Where isolation is desired for periodic inspection or maintenance of a relief device on a vessel required to operate continuously, the vessel may be fitted with an array of pressure relief devices and isolating valves so mechanically interlocked that the capacity of the relief devices remaining in service cannot under any conditions be reduced below that required by Clauses 8.2.1 and 8.2.2. Any such isolating valve shall be of the full-way type with port area not less than the area of the inlet of its associated relief device, and shall be of a type and so located that the obturator (e.g. valve disc) cannot be inadvertently disconnected and block the connection between the vessel and the relief device.

    Where agreed between the parties concerned a procedure of locking by authorized personnel may be used as an alternative to mechanical interlocking. Each isolating valve shall be capable of being locked or sealed in the open position.

(b) Where the pressurizing of a vessel can originate only from an outside source, an isolating valve may be fitted between the vessel and the relief device, provided that the same valve simultaneously isolates the vessel from the means of pressurizing and a suitable device to protect against excess pressure in the event of fire (see Clause 8.6.2), is directly fitted to the vessel without any means of isolation.

8.9 DISCHARGE FROM PRESSURE-RELIEF DEVICES

8.9.1 Safe discharge

The discharge from pressure-relief devices shall be made in such a manner as to prevent danger to persons, damage to equipment or environment and preferably to a place where the discharge is visible. Discharge to lower pressure systems is permissible provided that the receiving system can accept the additional load without causing an unacceptable back pressure.

8.9.2 Discharge to atmosphere

Unless otherwise provided in the relevant application code, toxic or flammable fluids (where agreed by the parties concerned and other relevant statutory authorities), and other fluids, may be discharged from a static vessel to atmosphere provided that the discharge is outside of and away from buildings, preferably through a vertical pipe to a height of at least 2 m above the vessel or the building in which the vessel is installed. All relief devices shall be arranged so that the discharge does not impinge on the vessel, and the cooling effect will not prevent effective operation of the device, e.g. vessels containing carbon dioxide or nitrous oxide.

8.9.3 Discharge pipes

Discharge pipes from pressure-relief valves shall be sized in accordance with AS 4041, so that, under maximum discharge conditions, the build-up of back pressure at the outlet of the valve as the result of the valve discharging does not reduce the relieving capacity of the valve below that required to protect the vessel.

The bore of the discharge pipe shall be not less than the bore of the outlet of the pressure-relief device.

Discharge piping shall run as directly as practicable to the point of final release.

Discharge piping shall be adequately and independently supported to prevent transmittal of forces due to the mass of the tube, discharge reaction and thermal expansion strain. Forces acting on a safety or relief valve should be kept to a minimum under all conditions of operation.

8.9.4 Common discharge pipes

Where it is not feasible to provide each pressure-relief device with a separate discharge pipe, a discharge pipe common to a number of devices on one or more vessels may be used by agreement of the parties concerned. In such installations where it is necessary to provide stop valves to permit relief valve maintenance—

(a) a valve or valving arrangement shall be connected to the outlet of the pressure-relief valve which shall be designed to connect the outlet of the valve to the atmosphere while disconnecting the outlet from the common discharge pipe and vice versa; and

(b) the valve or valving arrangement referred to in Item (a) shall meet the requirements of Clause 8.8.4.

The size of a common discharge pipe serving two or more pressure-relief devices that may reasonably be expected to discharge simultaneously, shall ensure that the required total discharge capacity can be achieved. The total pipe area should at least equal the sum of their outlet areas, with due allowance for pressure drop in the downstream sections. Pressure-relief valves specially designed for use on high or variable back pressure should be considered.

8.9.5 Drainage

In addition to the requirements of Clause 8.4.7, discharge pipes shall be designed to facilitate drainage or shall be fitted with an open drain to prevent liquid from lodging on the discharge side of the device. Precautions shall be taken to prevent rainwater from collecting in vertical discharge pipes.

8.9.6 Bonnet and pilot valve venting

The venting of the bonnet of valves and of pilot-operated valves, where required, shall also comply with the above requirements. Precautions shall be taken in the design of the vent piping to avoid any possibility of back pressure on the pilot.

8.9.7 Noise

Discharge from pressure relief devices may create excessive noise. Depending on frequency, duration of discharge and location, silencers may need to be fitted to discharge lines. Care shall be taken to ensure that they do not create an obstruction or excessive downstream pressure drop.

8.10 VACUUM-RELIEF DEVICES

8.10.1 Application

Where sub-atmospheric pressures may occur (including reduced pressure due to cooling of the contents) and the vessel is incapable of withstanding such conditions, a vacuum-relief device shall be fitted to prevent collapse.

8.10.2 Design, manufacture, testing and marking

The design, manufacture, testing and marking of vacuum-relief devices shall comply with the general requirements of AS 1271.

8.10.3 Required capacity and setting

8.10.3.1 *General*

The capacity and setting of the vacuum-relief device(s) shall be such as to provide the necessary rate of air (or gas) flow, so that the absolute pressure will not fall below that for which the vessel is designed.

8.10.3.2 *Sizing of vacuum breakers for feedwater deaerators*

The capacity of the vacuum relief devices installed on a feedwater deaerator shall be:

$$Q_a = \frac{V_s \Delta h m'}{L} \qquad \ldots 8.10.3.1$$

where

Q_a = minimum aggregate vapour capacity of relief devices, in cubic metres per minute, of air at the deaerator vacuum design condition

V_s = specific volume of steam at the deaerator vacuum design condition, in cubic metres per kilogram

Δh = change in enthalpy of the feedwater as it is heated from its inlet temperature condition to boiling point at the deaerator vacuum design condition, in joules per kilogram.

m' = maximum flow of feedwater into the deaerator, in kilograms per minute

L = latent heat of vaporization of steam at the deaerator vacuum design condition, in joules per kilogram

8.10.4 Installation

Vacuum-relief devices shall be installed in the same manner as pressure-relief devices (see Clauses 8.8 and 8.9), suitably amended for vacuum conditions. Particular care shall be exercised in the design and installation of the air inlet to such devices to prevent possible blockage.

8.11 FUSIBLE PLUGS

8.11.1 Definition

A fusible plug is an operating part, usually in the form of a plug of suitable low melting-point material (usually a metal alloy), which initially blocks a discharge opening in the vessel under normal conditions, but yields or melts at a predetermined temperature to discharge fluid for the relief of pressure.

8.11.2 Application

By agreement between the parties concerned, one or more fusible plugs may be used in lieu of pressure-relief devices only in special applications, e.g. to provide protection in the event of fire around a vessel which is isolated from a safety valve, and under the following conditions:

(a) A pressure-relief device is required only for the protection of the vessel in the event of a fire or other unexpected source of extreme heat.

(b) The service conditions and installation are such that deposits will neither shield the device (causing an increase in temperature necessary to fuse the plug) nor restrict the discharge.

(c) The contents of the vessel are non-toxic and non-flammable and the water capacity of the vessel does not exceed 500 L, or the content of the vessel is toxic or flammable and the water capacity of the vessel does not exceed 100 L.

(d) The plugs comply with the remaining requirements of this Clause (8.11).

In special instances and with agreement of the parties concerned, a soft brazed or soldered joint with appropriate yield temperature may be used in lieu of a fusible plug.

8.11.3 Design, manufacture, testing and marking

Plugs shall be in accordance with AS 2613.

8.11.4 Required discharge capacity

The minimum discharge capacity required to protect the vessel shall be determined in accordance with Clause 8.6.2 or, if appropriate, with AS 2613.

The size and number of fusible plugs shall be sufficient to relieve the above minimum discharge capacity.

8.11.5 Required yield temperature

Fusible plugs shall have a maximum yield temperature (i.e. a specified yield temperature plus 3°C) not exceeding the temperature which would result in a rise in vessel pressure to 120% of the design pressure of the vessel.

For vessels containing liquefied flammable or toxic gases at ambient temperatures, the specified yield temperature shall comply with the above requirement and shall be not less than 5°C above the temperature used as a basis for the design pressure.

For vessels containing permanent gases at ambient temperatures, the specified yield temperature shall not exceed 80°C, except that for air receivers using plugs for the protection required by Clause 8.11.2(d) the specified yield temperature shall not exceed 150°C. The specified yield temperature should not be less than 70°C.

8.11.6 Installation

Fusible plugs shall be connected to the vapour space and located in positions that will represent the highest temperature of the vessel and its contents.

Where the vessel length exceeds 750 mm, at least one fusible plug shall be installed at each end of the vessel and each shall have the full capacity required to protect the vessel.

The installation shall comply with Clause 8.9. Where the vessel is in a location where the collection of discharged gas would be dangerous, e.g. toxic or flammable or carbon dioxide gas, it is recommended that the discharge from the fusible plug be piped to atmosphere. The connection of piping shall be designed to minimize its influence on the yield temperature of the plug.

8.12 PROTECTION AGAINST OPERATION OUTSIDE DESIGN TEMPERATURE LIMITS

Where the temperature of a pressure-containing part of a vessel is outside the minimum or maximum design temperature limits while still subject to pressure (or where the maximum stress in a part may exceed the design strength for the temperature of the part) due to a credible failure of a single temperature, flow or level control device, consideration shall be given to the fitting of one or more additional safety devices that will limit the temperature and pressure to acceptable levels, or to the fitting of temperature actuated devices capable of relieving pressure (see Clause 8.11). Such safety devices are the subject of agreement between the parties concerned and should be independent of, and additional to, the single control device and of a fail-safe design. (See also Clause 8.6.2.) Rapid changes of temperature due to credible failures shall also be considered.

Pressure-relief devices might not protect a vessel from excessive temperature in a fire. For example, the wall temperature of an LP Gas vessel might reach a sufficiently high temperature to cause bursting at or below the maximum permitted accumulation. In critical localities consideration should be given to the use of—

(a) pressure-reducing systems, to avoid bursting of vessels containing lethal or flammable gases;

(b) water or foam spray systems, fire barriers, thermal insulation or separation, to limit metal temperature (insulation is advantageous in water-scarce areas and to reduce relieving capacity);

(c) fire suppressant foam or an inert gas system;

(d) low-stress design, to increase heat capacity and creep time, and so increase the time available for effective emergency action; or

(e) a combination of these measures.

8.13 PRESSURE GAUGES

8.13.1 Application

At least one pressure gauge shall be provided for each vessel fitted with a pressure-relief device, unless otherwise agreed between the parties concerned.

8.13.2 Type and size

Gauges shall comply with AS 1349 or other Standard agreed by the parties concerned. It is recommended that static vessels be fitted with a bourdon tube type pressure gauge and transportable vessels with a Schaffer or diaphragm type of gauge.

The nominal size shall be not less than 75 mm diameter except that in vessels less than 380 mm diameter, a 50 mm diameter gauge may be used. The operating pressure shall fall within the middle third of the graduated range of the gauge and a red line shall mark the operating pressure. Where a gauge is compensated for head of liquid between the gauge and the vessel connection, the amount of such compensation should be marked on the dial.

As an alternative digital type pressure gauges may be used provided they have clear readability, reliability and accuracy equivalent to AS 1349 and can be readily calibrated.

8.13.3 Connection

The pressure gauge should preferably be located on the vessel itself, but may be adjacent to the vessel on the inlet pipe. Where a number of vessels are connected to the same system, one gauge will suffice for all vessels provided that these vessels operate at the same pressure and the gauge may at all times be capable of connection so as to indicate the pressure at any relevant relieving device.

It is recommended that a shut-off valve be fitted between the vessel and the gauge, particularly where a vessel cannot be readily removed from service for replacement of the gauge.

Gauges shall be visible from the position where the operator controls the vessel pressure or opens quick-acting covers and shall be fitted with some device such as a syphon pipe to prevent excessive temperature reaching the operating element of the pressure gauge.

8.14 LIQUID LEVEL INDICATORS

8.14.1 General

Where liquid level indicators are required, the pressure retaining components of the indicators shall comply with the general design and manufacture requirements of AS 1271 (or other equivalent Standard) or this Standard (AS 1210) and the indicator shall be capable of indicating the liquid level to the required accuracy.

8.14.2 Tubular glass indicators

Tubular glass indicators shall comply with AS 1271 and all passageways shall be so constructed that a cleaning instrument can be passed through them. They shall be adequately protected against damage, and be suitably guarded to prevent injury to persons in the event of failure.

Tubular glass indicators shall not be used for lethal or toxic materials or on transportable vessels.

8.15 ISOLATION FITTINGS

Where it is necessary for inspection, maintenance or other purposes, suitable provision shall be made to isolate the vessel from all pressure sources.

Where the source of pressure is from another connected vessel which may be in operation during inspection, the provision for isolation shall be one of the following:

(a) One stop valve and a blanking plate.

(b) Two stop valves with an atmospheric vent between.

(c) Removal of a section of interconnecting pipework.

All provisions for isolation are located between the vessel and each connected source of pressure. Where the source of pressure serves only one vessel, one stop valve only is required provided the pressure source can be rendered inoperative.

NOTE: Where other valves are fitted and comply with the requirements of Clause 8.8.4, such valves may be considered for compliance with the requirements of (a) or (b) above. Where such valves comply with the requirements of Clause 8.15, no further valves should be necessary.

8.16 DRAINAGE

8.16.1 Provision for drainage

Unless otherwise permitted in the relevant application code, provision shall be made for the complete drainage of a vessel that contains or is likely to contain material which is corrosive to the vessel (e.g. water in air receivers) or which is toxic or flammable. This may involve a suitable drain located at the lowest part of the vessel and a full-way valve. The size of the valve shall not be less than 10 mm. The size of the valve should be at least 20 mm.

8.16.2 Discharge

Where a drain valve is required to discharge toxic or flammable material, discharge piping shall be attached to the valve and shall lead to a safe location. The discharge shall be made in a manner to prevent danger to persons or damage to equipment and environment and preferably so that the discharge is visible.

8.17 VENTS

Provision shall be made to vent air from the highest parts of the vessel during hydrostatic test. Where openings provided for other purposes are not suitable, openings shall be provided and shall be sealed by any suitable means after testing.

8.18 PROTECTION OF VALVES AND FITTINGS

8.18.1 Location for inspection and maintenance

Pressure-relief devices, other safety devices and important vessel fittings shall be located and installed so that they are readily accessible for operation, inspection, maintenance and removal.

8.18.2 Protection against interference

Where the pressure setting or other adjustment is external to the safety device, the adjustment shall be locked and/or sealed, unless otherwise agreed between the parties concerned. Such devices and fittings shall be installed and protected so that they cannot be readily rendered inoperative or be tampered with, and so that entrance of dirt, water, wildlife and other deleterious material to the valve outlet will be at a minimum. Devices shall be protected or located to prevent freezing from making the device inoperative.

8.18.3 Protection against damage

All safety valves and fittings on vessels shall be arranged, where possible, to afford maximum protection against accidental damage. See Clause 3.26 for protection in relation to transportable vessels.

SECTION 9 PROVISIONS FOR DISPATCH

9.1 CLEANING

On completion of the vessel and prior to dispatch, all vessels shall be cleaned and shall be free from loose scale and other foreign matter. (See AS 4458.)

NOTE: Specific requirements for cleaning and surface treatment should be agreed between the purchaser and manufacturer.

9.2 PROTECTION

Prior to dispatch, the vessel shall be protected as necessary against damage during transport and storage prior to erection. The extent and responsibility for such protection is the subject of agreement between the purchaser and the manufacturer and should have due regard to physical damage and/or corrosion, which may occur due to the method and conditions of transport and storage, and the time that may elapse before erection.

Particular attention shall be given to the protection of machined surfaces, and the compatibility therewith, as regards corrosion, of any material used for physical protection. The possibility of distortion of the vessel and any of its parts shall receive special consideration. Where appropriate, suitable provision shall be made for lifting, supporting and anchoring the vessel.

9.3 ASSOCIATED FITTINGS AND COMPONENTS

Arrangement for the protection or the separate supply of the vessel's protective devices and associated fittings is the subject of agreement between purchaser and manufacturer where applicable.

SECTION 10 NON-METALLIC VESSELS

10.1 SCOPE

Previous Sections of this Standard deal specifically with metallic vessels. This Section applies to pressure vessels or vessel pressure parts made of plastics, fibre-reinforced plastics, glass or other non-metallic materials, except for gaskets (see Clause 3.21.5.1).

10.2 GENERAL REQUIREMENTS

Non-metallic vessels should comply with the general principles of this Standard. They shall comply with the following:

(a) The following Sections and Clauses that apply to all vessels:

Section 1—except in Clause 1.6 a suitable classification system shall identify the type of construction.

Section 2—Clause 2.1 only, with the requirement that all materials shall be suitable for all anticipated service conditions. Allowance shall be made for any ageing or embrittlement and for suitable performance under foreseeable fire conditions (with or without protection).

Section 3—Clauses 3.1, 3.2, 3.3, 3.4 and 3.26 as appropriate.

Section 4—Does not apply, see Item (b).

Section 5—Applicable Clauses only.

Section 6—All Clauses apply.

Section 7—Equivalent marks and reports.

Section 8—Applicable Clauses only.

Section 9—All Clauses apply.

(b) ASME BPV-X, BS 4994, AS 2971 or other national Standard agreed by parties concerned but within the limits of each Standard.

(c) The specific engineering design and any applicable regulations.

(d) All conditions agreed by the parties concerned.

Materials referenced in the Standards above may be substituted by an equivalent Australian or overseas material specification.

NOTE: Materials should be used within the conditions recommended by the material manufacturer.

APPENDIX A

BASIS OF DESIGN TENSILE STRENGTH

(Normative)

A1 GENERAL

Table A1 describes the derivation of design tensile strengths for this Standard.

The stresses given in Table B1 (and Table B2 for bolting) are the design tensile strengths, f, and are intended to be interpreted as the maximum allowable primary membrane stresses. They are based on the criteria given below using mechanical properties as indicated, with some exceptions.

In some instances the design strengths listed are applicable to a limited range of thicknesses. This has been done to simplify presentation where specified properties vary with range of thickness. For design strengths beyond these thickness limits, the material data in the material specification shall be used.

These design strengths do not include an allowance for weld joint efficiency (η) (except for welded tubes complying with ASTM specifications) or casting quality factor, as these are provided for in the appropriate clauses of this Standard. The values are for materials in the non-post weld heat treated condition.

For materials not listed in the Tables in Appendix B, the design strength may be taken from one of the following:

(a) ASME BPV-VIII-1 for Classes 1, 2A, 2B and 3, excluding grade ASME SA/AS1548.

(b) ASME BPV-VIII-2 for a conservative approximation to Classes 1H and 2H, excluding grade ASME SA/AS1548.

(c) PD 5500 for Classes 1H and 2H.

(d) As described in Paragraph A3.

NOTES:

1 ASME BPV-II-D lists design strengths that do not comply with the requirements of this Standard for bolting materials. This Standard uses a more conservative design strength basis for bolting materials in order to avoid bolt deformation and joint leakage in service.

2 Design strengths for AS 1548 grades listed in ASME BPV-II-D do not meet the requirements of this Standard. ASME design strengths are based on derived material properties that do not accurately reflect the performance of steels complying with AS 1548.

A2 NOTATION

The notation used for material properties is as follows:

A_5 = Specified minimum percentage elongation on gauge length = $5.65\sqrt{S_o}$ or $5d$

where

S_o = original cross-sectional area, in millimetres

d = original diameter, in millimetres

NOTE: Where the minimum elongation is not available, it may be approximated by half the reduction area.

R_m	=	Specified minimum tensile strength (approximating 3 standard deviations below the mean) for the grade of material concerned at room temperature (tested in accordance with AS 1391 or equivalent); or
	=	0.5 (R_m + R_{mACT}) where agreed by the designer and the manufacturer, and where R_{mACT} is actual tensile strength of each item of material used in the vessel, as recorded on the material maker's material test certificate. In this case the value of R_{mACT} is also limited to 125% of R_m, and suitable tests are required to assess the actual strength of formed ends or other parts where fabrication methods may reduce tensile strength.
	=	For iron castings, minimum tensile strength of test bars of sections appropriate to the vessel thickness.
R_{mT}	=	Specified minimum tensile strength for the grade of material concerned at design temperature T (tested in accordance with AS 2291 or equivalent) or for steels that have not been tested, is taken as equal to R_m, up to 350°C.
R_e	=	Specified minimum upper yield strength (R_{eH}, approximating 3 standard deviations below the mean) for the grade of material concerned at room temperature (tested in accordance with AS 1391 or equivalent); or
	=	0.5 (R_e + R_{eACT}) where agreed by the designer and the manufacturer, and where R_{eACT} is the actual yield strength of each item of material used in the vessel, as recorded on the material maker's material certificate. In this case suitable tests are required to assess the actual strength of formed ends or other parts where fabrication methods may reduce yield strength.
		Where a material Standard specifies minimum values of lower yield strength (R_{eL}) or proof strength (0.2% proof ($R_{p0.2}$); 1.0% proof strength for austenitic steels ($R_{p1.0}$) or $R_{t0.5}$ (for copper alloys), these values are taken as corresponding to R_e.
R_{eT}	=	Specified minimum value of R_{eL} or $R_{p0.2}$ ($R_{p1.0}$ for austenitic steels) for the grade of material concerned at design temperature T (tested in accordance with AS 2291 or equivalent), approximating 2 standard deviations below the mean.
S_R	=	Estimated mean stress to cause rupture in 100 000 h at the design temperature T for the grade of material concerned; if the width of the scatter band of test results exceeds ±20 percent of the mean value, then S_R shall be taken as 1.25 times the minimum rupture stress
S_{Rt}	=	Estimated mean stress to cause rupture in time t at the design temperature T for the grade of material concerned
S_c'	=	Estimated mean stress to cause an elongation (creep) of one percent in 100 000 h at design temperature T for the grade of material concerned.
S_{ct}	=	Estimated mean stress to cause an elongation (creep) of one percent in t hours at design temperature T for the grade of material concerned.

A3 DETERMINATION OF DESIGN TENSILE STRENGTH

The design tensile strength may be derived from Table A1, using material properties specified in—

(a) Australian material Standards;

(b) International pressure equipment material Standards (e.g. ISO 9328 series);

(c) ASME BPV-II-D (for ASME or ASTM materials) excluding grade ASME SA/AS1548; or

(d) an authoritative source using values for R_{eT} and R_{mT}, approximating 2 standard deviations below the mean.

The design tensile strength shall not exceed the lowest of the values according to Table A1 (see Note 1 to Table A1).

NOTES:

1 Design strengths for AS 1548 grades listed in ASME BPV-II-D do not meet the requirements of this Standard. ASME design strengths are based, on derived material properties that do not accurately reflect the performance of steels complying with AS 1548.

2 In the creep range, design strengths will be identical for Classes 1, 1H, 2A, 2B, 2H and 3.

TABLE A1

DETERMINATION OF DESIGN TENSILE STRENGTH

Material		Below creep range	In creep range
1	Vessels—Classes 1, 2A, 2B, 3		
1.1	All metals other than structural quality and bolting (Note 4)	$\frac{R_m}{3.5}, \frac{R_{mT}}{3.5}, \frac{R_e}{1.5}, \frac{R_{eT}}{1.5}$, and (for A_5 <15%) R_m (0.1 + 0.023 A_5)	For indefinite creep life, lowest of— $\frac{S_R}{1.5}$ and S'_c For creep life of t hours, lowest of— $\frac{S_{Rt}}{1.3}$ and S_{ct} (see Note 3)
1.2	Structural quality metals	Values determined in Item 1.1, multiplied by 0.92 (See Clauses 2.3.3 and 2.3.4)	Not permitted
2	Vessels—Classes 1H, 2H, 1S, 2S		
2.1	All metals other than structural quality and bolting (Note 4)	$\frac{R_m}{2.35}, \frac{R_{mT}}{2.35}$, (or $\frac{R_m}{2.5}, \frac{R_{mT}}{2.5}$ for austenitic steels, (see Note 2), $\frac{R_e}{1.5}, \frac{R_{eT}}{1.5}$, and (for A_5 <15%) R_m (0.1 + 0.023 A_5)	For indefinite creep life, lowest of— $\frac{S_R}{1.5}$ and S'_c For creep life t hours, lowest of— $\frac{S_{Rt}}{1.3}$ and S_{ct} (see Note 3)
2.2	Structural quality	Values determined in Item 2.1, multiplied by 0.92 (See Clauses 2.3.3 and 2.3.4)	Not permitted
3	Bolting for all classes	$\frac{R_m}{5}, \frac{R_{mT}}{4}, \frac{R_e}{4}, \frac{R_{eT}}{3}$	$\frac{S_R}{1.5}$; S'_c
4	Non-metallic vessels	For plastics, glass, graphite, or other non-metallic vessels, the design stresses are the subject of agreement between parties concerned with the provision that the design complies with the relevant vessel design standards.	

NOTES:

1 See Clause 3.33 for limits on use of R_{eT} and R_{mT} as alternatives to R_e and R_m for low temperature applications.

2 For 1S and 2S vessels, see Clause L3.3 for applicable design strengths.

3 See Clause 3.26.8 for application of S_R 600°C in 10 minutes, i.e. for transportable vessels Classes 1H and 1S using high strength material or where fire protection is needed.

4 Iron castings (i.e. grey cast iron, whiteheart and blackheart) are not permitted in the creep range.

5 Different safety factors are necessary for vessels designed for use as ISO tank containers to IMDG rules.

6 See Clause 3.26 for safety factors for transportable vessels.

APPENDIX B

MATERIAL DESIGN DATA

(Normative)

B1 DESIGN STRENGTHS—MATERIALS

A1 Tables B1(A) to (J) give design strength (*f*) values for a range of materials for use in vessel design. Table B1(A) gives design strength values for Classes 1H and 2H designs where $R_m/2.35$. Tables B1(B) to (H) give design strength values at $R_m/3.5$, for Classes 1, 2A, 2B and 3 vessels.

For material types and grades not listed, the design strengths may be determined according to Appendix A.

B2 DESIGN STRENGTHS—BOLTING

Table B2 gives design strength (*f*) values for bolting materials, for use with designs to Clause 3.21.

B3 YOUNG'S MODULUS

Table B3 gives values for Young's Modulus (*E*).

The data should not imply the materials are suitable at all temperatures listed. Linear interpolation of values is acceptable.

B4 LINEAR THERMAL EXPANSION

Table B4 lists the linear expansion from 20°C to the indicated temperature. For other materials, properties may be obtained from the manufacturer or another authoritative source.

These data should not imply the materials are suitable at all temperatures listed. Linear interpolation of values is acceptable.

TABLE B1(A)

DESIGN TENSILE STRENGTH FOR CLASS 1H AND 2H VESSELS (MPa)
(A) CARBON, CARBON-MANGANESE AND LOW ALLOY STEELS

Specification (Type)	Grade	Thickness mm	Steel group	R_m MPa	Notes	Design tensile strength, MPa (See Notes)													
						Temperature, °C													
						50	100	150	200	250	300	325	350	375	400	425	450	475	500
AS 1548	PT430	3≤t≤16	A1	430	—	**185**	**185**	173	162	151	140	135	130	127	122	119	—	—	—
(C, C-Mn)		16<t≤40				185	174	161	151	141	131	126	121	118	114	111	—	—	—
		40<t≤80				178	168	156	146	136	126	128	117	114	110	107	—	—	—
		80<t≤150				165	156	144	135	126	117	113	108	106	102	99	96	—	—
		All				Creep Life: Indefinite								—	*115*	*79*	*52*	*35*	—
						10 000 hrs								—	*179*	*141*	*105*	*74*	*52*
						30 000 hrs								—	*158*	*118*	*82*	*56*	*39*
						100 000 hrs								—	*133*	*91*	*60*	*40*	—
						150 000 hrs								—	*123*	*82*	*53*	*33*	—
						250 000 hrs								—	*114*	*73*	*48*	—	—
	PT460	8≤t≤16	A1	460	—	**198**	189	176	165	150	143	138	132	128	124	120	116	—	—
		16<t≤40				195	183	170	159	148	138	133	128	124	120	116	112	—	—
		40<t≤80				182	170	158	148	138	128	124	119	116	112	108	104	—	—
		80<t≤150				175	165	152	142	133	124	120	115	111	107	99	—	—	—
		All				Creep Life: Indefinite								—	*115*	*79*	*52*	*35*	—
						10 000 hrs								—	*179*	*141*	*105*	*74*	*52*
						30 000 hrs								—	*158*	*118*	*82*	*56*	*39*
						100 000 hrs								—	*133*	*91*	*60*	*40*	—
						150 000 hrs								—	*123*	*82*	*53*	*33*	—
						250 000 hrs								—	*114*	*73*	*48*	—	—
	PT490	8≤t≤16	A2	490	—	211	211	207	194	181	169	163	156	151	146	—	—	—	—
		16<t≤40				211	211	196	183	171	159	153	147	142	138	135	—	—	—
		40<t≤80				211	205	190	178	166	155	149	143	138	134	130	—	—	—
		80<t≤150				211	199	183	172	161	151	145	138	135	130	127	—	—	—
						Creep Life: Indefinite								145	110	*76*	*49*	*33*	—
						10 000 hrs								211	171	*135*	*100*	*69*	50
						30 000 hrs								195	151	*113*	*78*	*57*	38
						100 000 hrs								167	127	*87*	*57*	*38*	—
						150 000 hrs								160	117	*78*	*51*	*33*	—
	PT540	8≤t≤16	A3	540	—	**233**	**233**	**233**	**233**	218	202	—	—	—	—	—	—	—	—
		16<t≤40				**233**	**233**	**233**	222	204	189	—	—	—	—	—	—	—	—
AS 3597	700 PV	5<t≤25	G	790	—	**336**	**328**	**323**	**321**	**317**	**313**	**306**	**298**	—	—	—	—	—	—
(Low alloy		25<t≤65				**336**	**326**	**321**	**317**	**313**	**309**	**303**	**298**	—	—	—	—	—	—
Q&T)		65<t≤110	G	720		**306**	**294**	**291**	**287**	**285**	**283**	**280**	**277**	—	—	—	—	—	—

NOTES TO TABLE B1(A):

1 Design strengths at intermediate temperatures may be obtained by linear interpolation. Joint efficiencies and casting quality factors are to be applied where applicable. See Clause 3.3.1.

2 Design strengths in the creep range shall be the lesser of the non-creep value (shown in shaded cells) and the value for the relevant creep life (values shown in italics).

3 Values shown in bold are based on R_m/2.35, and require adjustment where other safety factors apply.

4 Values are for plate without post-weld heat treatment.

5 For design strengths at temperatures below 50°C, see Clause 3.3.2.

TABLE B1(B)

DESIGN TENSILE STRENGTH FOR CLASS 1, 2A, 2B AND 3 VESSELS (MPa)
(B) CARBON, CARBON-MANGANESE AND LOW ALLOY STEELS

Type	Specification (Type)	Grades	Thickness mm	Steel group (R_m, MPa)	Notes	Design tensile strength, MPa (Notes 1,2,3,4,5) Temperature, °C																			
						50	100	150	200	250	300	325	350	375	400	425	450	475	500	525	550	575	600	625	650
PLATE, SHEET AND STRIP																									
C, C-Mn	AS 1548	PT430	3≤t≤16	A1	—	**124**	**124**	**124**	**124**	**124**	**124**	**124**	**124**	**124**	122	119	—	—	—	—	—	—	—	—	—
			16<t≤40	(430)		**124**	**124**	**124**	**124**	**124**	**124**	**124**	121	118	114	111	—	—	—	—	—	—	—	—	—
			40<t≤80			**124**	**124**	**124**	**124**	**124**	**124**	**124**	117	114	110	107	—	—	—	—	—	—	—	—	—
			80<t≤150			**124**	**124**	**124**	**124**	**124**	117	113	108	106	102	99	96	—	—	—	—	—	—	—	—
			All			Creep Life: Indefinite								—	*115*	*79*	*52*	*35*	—	—	—	—	—	—	—
						10 000 hrs								—	*179*	*141*	*105*	*74*	*52*	—	—	—	—	—	—
						30 000 hrs								—	*158*	*118*	*82*	*56*	*39*	—	—	—	—	—	—
						100 000 hrs								—	*133*	*91*	*60*	*40*	—	—	—	—	—	—	—
						150 000 hrs								—	*123*	*82*	*53*	*33*	—	—	—	—	—	—	—
						250 000 hrs								—	*114*	*73*	*48*	—	—	—	—	—	—	—	—
		PT460	8≤t≤16	A1	—	**133**	**133**	**133**	**133**	**133**	**133**	**133**	132	128	124	120	116	—	—	—	—	—	—	—	—
			16<t≤40	(460)		**133**	**133**	**133**	**133**	**133**	**133**	133	128	124	120	116	112	—	—	—	—	—	—	—	—
			40<t≤80			**133**	**133**	**133**	**133**	**133**	128	124	119	116	112	108	104	—	—	—	—	—	—	—	—
			80<t≤150			**133**	**133**	**133**	**133**	**133**	124	120	115	111	107	99	—	—	—	—	—	—	—	—	—
			All			Creep Life: Indefinite								—	*115*	*79*	*52*	*35*	—	—	—	—	—	—	—
						10 000 hrs								—	*179*	*141*	*105*	*74*	*52*	—	—	—	—	—	—
						30 000 hrs								—	*158*	*118*	*82*	*56*	*39*	—	—	—	—	—	—
						100 000 hrs								—	*133*	*91*	*60*	*40*	—	—	—	—	—	—	—
						150 000 hrs								—	*123*	*82*	*53*	*33*	—	—	—	—	—	—	—
						250 000 hrs								—	*114*	*73*	*48*	—	—	—	—	—	—	—	—
		PT490	8≤t≤16	A2	—	**142**	**142**	**142**	**142**	**142**	**142**	**142**	**142**	**142**	**142**	—	—	—	—	—	—	—	—	—	—
			16<t≤40	(490)		**142**	**142**	**142**	**142**	**142**	**142**	**142**	**142**	**142**	138	135	—	—	—	—	—	—	—	—	—
			40<t≤80			**142**	**142**	**142**	**142**	**142**	**142**	**142**	**142**	138	134	130	—	—	—	—	—	—	—	—	—
			80<t≤150			**142**	**142**	**142**	**142**	**142**	**142**	**142**	138	135	130	127	—	—	—	—	—	—	—	—	—
			All			Creep Life: Indefinite								*145*	*110*	*76*	*49*	*33*	—	—	—	—	—	—	—
						10 000 hrs								*211*	*171*	*135*	*100*	*69*	*50*	—	—	—	—	—	—
						30 000 hrs								*195*	*151*	*113*	*78*	*57*	*38*	—	—	—	—	—	—
						100 000 hrs								*167*	*127*	*87*	*57*	*38*	—	—	—	—	—	—	—
						150 000 hrs								*160*	*117*	*78*	*51*	*33*	—	—	—	—	—	—	—
		PT540	8≤t≤16	A3	—	**156**	**156**	**156**	**156**	**156**	**156**	—	—	—	—	—	—	—	—	—	—	—	—	—	—
			16<t≤40	(540)		**156**	**156**	**156**	**156**	**156**	**156**	—	—	—	—	—	—	—	—	—	—	—	—	—	—

NOTE: See end of this Table.

(*continued*)

TABLE B1(B) (*continued*)

Type	Specification (Type)	Grades	Thickness mm	Steel group (R_m, MPa)	Notes	Design tensile strength, MPa (Notes 1,2,3,4,5)																			
						Temperature, °C																			
						50	100	150	200	250	300	325	350	375	400	425	450	475	500	525	550	575	600	625	650
PLATE, SHEET AND STRIP (*continued*)																									
	AS/NZS 3678	250	All	A1 (410)	6, 7	108	108	108	106	99	93	90	86	—	—	—	—	—	—	—	—	—	—	—	—
		300	All	A1 (430)		113	113	113	113	113	113	109	105	—	—	—	—	—	—	—	—	—	—	—	—
		350	All	A2 (450)		118	118	118	118	118	118	118	118	—	—	—	—	—	—	—	—	—	—	—	—
		400	All	A2 (480)		126	126	126	126	126	126	126	126	—	—	—	—	—	—	—	—	—	—	—	—
	AS/NZS 1594	HA200	≤16	A1 (300)	7	86	86	86	86	86	86	86	81	—	—	—	—	—	—	—	—	—	—	—	—
		HU,HA250	≤16	A1 (350)		100	100	100	100	100	100	100	100	—	—	—	—	—	—	—	—	—	—	—	—
		HU,HA300	≤16	A1 (400)		114	114	114	114	114	114	114	114	—	—	—	—	—	—	—	—	—	—	—	—
		HA300/1	≤16	A1 (430)		123	123	123	123	123	123	123	121	—	—	—	—	—	—	—	—	—	—	—	—
		HA350	≤16	A3 (430)		123	123	123	123	123	123	123	123	—	—	—	—	—	—	—	—	—	—	—	—
		XF300	≤8	A1 (440)		126	126	126	126	126	126	123	121	—	—	—	—	—	—	—	—	—	—	—	—
		HA400	≤16	A3 (460)		131	131	131	131	131	131	131	131	—	—	—	—	—	—	—	—	—	—	—	—
		XF400	≤8	A3 (460)		131	131	131	131	131	131	131	131	—	—	—	—	—	—	—	—	—	—	—	—
		XF500	≤8	A3 (570)		163	163	163	163	163	163	163	163	—	—	—	—	—	—	—	—	—	—	—	—
Low Alloy Q&T	AS 3597	700 PV	5≤t≤65	G (720)	—	226	226	226	226	226	226	226	226	—	—	—	—	—	—	—	—	—	—	—	—
			>65	G (720)		206	206	206	206	206	206	206	206	—	—	—	—	—	—	—	—	—	—	—	—
C-½Mo	ASTM A 204	A	All	B	—	128	128	128	128	128	128	128	128	128	128	127	124	105	69	43	23	—	—	—	—
		B	All	B	—	138	138	138	138	138	138	138	138	138	138	137	134	107	68	43	23	—	—	—	—
		C	All	B	—	148	148	148	148	148	148	148	148	148	148	148	144	109	67	43	23	—	—	—	—
½Cr-½Mo	ASTM A 387	2 Class 1	All	D1	—	108	108	108	108	108	108	108	108	108	108	108	106	103	77	50	31	—	—	—	—
		2 Class 2	All	D1	—	138	138	138	138	138	138	138	138	138	138	138	135	130	89	46	33	—	—	—	—
1Cr-½Mo	ASTM A 387	12 Class 1	All	D2	—	108	106	104	104	104	104	104	104	104	104	104	104	102	89	62	40	26	17	12	7
		12 Class 2	All	D2	—	128	125	123	123	123	123	123	123	123	123	123	123	123	94	60	41	26	17	12	7
1¼Cr-½Mo-Si	ASTM A 387	11 Class 1	All	D2	—	118	118	118	118	118	118	118	118	118	118	116	114	101	75	52	37	25	18	12	8
	ASTM A 387	11 Class 2	All	D2	—	148	148	148	148	148	148	148	148	148	148	148	143	107	73	52	36	25	18	12	8
2¼Cr-1Mo	ASTM A 387	22 Class 1	All	D2	—	118	118	114	114	114	114	114	114	114	114	114	114	100	81	64	48	35	24	16	9
	ASTM A 387	22 Class 2	All	D2	—	148	147	144	142	141	141	140	139	138	136	133	130	116	89	64	45	30	20	13	8
3Cr-1Mo	ASTM A 387	21 Class 1	All	D2	—	118	118	114	114	114	114	114	114	114	114	114	113	91	68	54	44	34	25	17	10
	ASTM A 387	21 Class 2	All	D2	—	148	147	144	142	141	141	140	139	138	136	133	127	100	73	55	41	29	20	16	9
5Cr-½Mo	ASTM A 387	5 Class 1	All	D2	—	118	118	114	114	113	112	111	109	107	104	100	96	81	62	46	35	26	18	12	7
	ASTM A 387	5 Class 2	All	D2	—	148	147	143	142	142	140	139	137	134	130	126	104	81	62	46	35	26	18	12	7
9Cr-1Mo-V	ASTM A 387	91 Class 2	t ≤75	D2	—	168	168	168	167	166	164	163	161	157	153	147	141	134	126	117	107	89	65	46	29
	ASTM A 387	91 Class 2	t >75	D2	—	168	168	168	167	166	164	163	161	157	153	147	141	134	126	118	103	81	62	46	29

NOTE: See end of this Table.

(*continued*)

TABLE B1(B) (*continued*)

Type	Specification (Type)	Grades	Thickness mm	Steel group (R_m, MPa)	Notes	Design tensile strength, MPa (Notes 5, 8) Temperature, °C 50	100	150	200	250	300	325	350	375	400	425	450	475	500	525	550	575	600	625	650
PIPE AND TUBE																									
Carbon	ASTM A 53	E/A	All	—	—	81	81	81	81	81	81	81	80	72	62	55	48	40	33	—	—	—	—	—	—
		S/A	All		—	95	95	95	95	95	95	95	95	84	73	65	56	48	36	—	—	—	—	—	—
	ASTM A 106	A	All		—	95	95	95	95	95	95	95	94	84	73	65	56	48	36	24	11	—	—	—	—
C-Mn	ASTM A 53	E/B	All	—	—	101	101	101	101	101	101	101	99	90	76	64	53	39	27	—	—	—	—	—	—
		S/B	All		—	118	118	118	118	118	118	118	117	105	89	75	63	46	32	—	—	—	—	—	—
	ASTM A 106	B	All		—	118	118	118	118	118	118	118	117	105	89	75	63	46	32	22	13	—	—	—	—
		C	All		—	138	138	138	138	138	138	138	135	123	101	84	67	51	34	21	13	—	—	—	—
C,C-Mn & low alloy steel	Pressure equipment types	—	—	A to G	Use design values determined from Appendix A, Paragraph A3. Values from AS 4041 may be used except where they exceed (R_m/3.5).																				
CASTING																									
Carbon	ASTM A 216	WCA	All	A1	—	118	118	118	118	114	107	104	101	98	89	75	63	46	32	22	13	—	—	—	—
		WCB	All	A2	—	138	138	138	138	136	129	125	122	117	101	84	67	51	34	21	13	—	—	—	—
		WCC	All	A2	—	138	138	138	138	138	138	138	135	123	101	84	67	51	34	21	13	—	—	—	—
FORGINGS																									
Carbon	ASTM A 105	—	All	A2	—	138	138	138	138	136	129	125	122	117	101	84	67	51	34	21	13	—	—	—	—
	ASTM A 266	1	—	A1	—	118	118	118	118	114	107	104	101	98	89	75	63	46	32	22	13	—	—	—	—
		2		A2	—	138	138	138	138	136	129	125	122	117	101	84	67	51	34	21	13	—	—	—	—
		4		A2	—	138	138	138	138	136	129	125	122	117	101	84	67	51	34	21	13	—	—	—	—
		3		A2	8	118	118	118	118	114	107	104	101	98	89	75	63	46	32	22	13	—	—	—	—
	ASTM A 350	LF2-1	—	A1	—	138	138	138	138	136	129	125	122	117	101	84	67	51	34	21	13	—	—	—	—
		LF2-2	—	A2	—	138	138	138	138	136	129	125	122	117	101	84	67	51	34	21	13	—	—	—	—
SECTION AND BARS (Note 12)																									
C, C-Mn	AS/NZS 3679.1	250	—	A1 (410)	7	108	108	108	106	99	93	90	86	—	—	—	—	—	—	—	—	—	—	—	—
		300	—	A1 (440)	7	116	116	116	116	116	116	116	111	—	—	—	—	—	—	—	—	—	—	—	—
		350	—	A2 (480)	7	126	126	126	126	126	126	126	126	—	—	—	—	—	—	—	—	—	—	—	—

NOTES:

1 Design strengths at intermediate temperatures may be obtained by linear interpolation. Joint efficiencies and casting quality factors are to be applied where applicable. See Clause 3.3.1.

2 Design strengths in the creep range shall be the lesser of the non-creep value (shown in shaded cells) and the value for the relevant creep life (values shown in italics).

3 Values shown in bold are based on R_m/3.5, and require adjustment where other safety factors apply.

4 Values are for plate without post-weld heat treatment.

5 For design strengths at temperatures below 50°C, see Clause 3.3.2.

6 Hot forming above 650°C, or normalizing of AS/NZS 1594 and AS/NZS 3678 steels is not to be performed unless the specified properties of the material are verified by tests on a specimen subject to a simulated heat treatment equivalent to that to which the vessel is subjected.

7 Thickness to satisfy requirements of Clause 2.3.3; see also Clause 2.3.4.

8 Welding and oxygen or other thermal cutting processes are not permitted when carbon content exceeds 0.35% by heat analysis.

A2

TABLE B1(C)

DESIGN TENSILE STRENGTH (MPa) HIGH ALLOY STEEL

ASTM spec.	Type or grade	Nominal composition	Steel group	Design tensile strength, MPa — Temperature, °C														
				50	100	150	200	250	300	350	400	450	500	550	600	650	700	750
A240	304	18Cr-8Ni	K	148	137	130	126	122	116	111	107	103	99.3	93.3	65	41.7	26.5	11.1
A240	304L	18Cr-8Ni	K	136	117	115	110	103	97.7	94.1	91.3	88.7	—	—	—	—	—	—
A240	316	16Cr-12Ni-2Mo	K	148	139	138	134	126	119	114	111	108	107	105	80	50.4	29.6	17.7
A240	316L	16Cr-12Ni-2Mo	K	138	116	115	109	103	98	94.1	90.9	87.8	—	—	—	—	—	—
A240	347	18Cr-10Ni-Cb	K	148	148	139	131	125	120	116	116	116	115	100	58	30.0	16.3	8.9
A240	S31008	—	—	148	142	138	138	135	129	125	122	119	112	59	32	16.9	6.1	2.4
A240	S31803	22Cr-5Ni-Mo-N	M	177	177	171	165	161	160	—	—	—	—	—	—	—	—	—
A240	S32101	—	—	186	169	160	154	154	—	—	—	—	—	—	—	—	—	—
A240	S32304	23Cr-4Ni-Mo-Cu	M	172	164	155	150	147	145	—	—	—	—	—	—	—	—	—
A240	S32205	—	—	187	176	171	165	161	160	—	—	—	—	—	—	—	—	—
A240	S32750	—	—	228	227	215	208	205	203	—	—	—	—	—	—	—	—	—
A240	S32906	—	—	215	213	204	198	196	195	—	—	—	—	—	—	—	—	—

NOTES:

1 These design strength values do not include a weld joint efficiency.

2 The design strength values in this Table may be interpolated to determine values for intermediate temperatures.

3 For design strengths at temperatures below 50°C, see Clause 3.3.2.

4 The above strength values used for A240 plate may also be used for forgings, seamless pipe, bars and other product forms that have no welds or other strength reduction characteristics. For welded pipe and castings the tabulated values for the relevant grade shall be multiplied by the weld joint efficiency or casting quality factor as appropriate.

TABLE B1(D)

DESIGN TENSILE STRENGTH (MPa) IRON CASTINGS

Material			Maximum allowable design pressure MPa	Notes	Design tensile strength, MPa							
Type	Specification	Grade			Temperature ≤ 250°C	Maximum thickness, mm (Note 1)						
						2	5	10	20	40	80	150
Grey iron	AS 1830 and ISO 185	JL/100	(Note 2)	1,3,4,5	—	11	11	11	11	11	—	—
		JL/150	(Note 2)	1,3,4	—	16	16	16	13	12	12	11
		JL/200	(Note 2)	1,3,4	—	21	21	21	18	18	16	15
		JL/225	(Note 2)	1,3,4	—	24	24	24	20	18	17	17
		JL/250	(Note 2)	1,3,4	—	27	27	27	22	20	18	18
		JL/275	(Note 2)	1,3,4	—	25	25	25	25	22	20	19
		JL/300	(Note 2)	1,3,4	—	32	32	32	27	24	22	22
		JL/350	(Note 2)	1,3,4	—	37	37	37	31	28	25	25
Spheroidal graphite iron	AS 1831 and ISO 1083	JS/350-22/U	(Note 6)	3,4,9	—	100	100	100	100	94	91	91
		JS/400-15/U	(Note 6)	3,4,9	—	114	114	114	114	111	106	106
		JS/400/18/U	(Note 6)	3,4,9	—	114	114	114	114	111	106	106
Whiteheart malleable iron	AS 1832 and ISO 5922	JMW/350-4	(Note 2)	3,4,7	—	27	31	35	—	—	—	—
		JMW/400-5	(Note 2)	3,4,7	—	30	36	40	—	—	—	—
Blackheart malleable iron	AS 1832 and ISO 5922	JMB/300-6	(Note 2)	3,4,8	—	30	30	30	30	—	—	—
		JMB/350-10	(Note 2)	3,4,8	—	35	35	35	35	—	—	—

(*continued*)

TABLE B1(D) (*continued*)

Material			Maximum allowable design pressure MPa	Notes	Design tensile strength, MPa							
Type	Specification	Grade			Temperature ≤ 250°C	Maximum thickness, mm (Note 1)						
						2	5	10	20	40	80	150
Spheroidal graphite austenitic iron	AS 1833 and ISO 2892	S-Ni Mn 13 7	(Note 6)	3,4	111	—	—	—	—	—	—	—
		S-Ni Cr 20 2	(Note 2)	3,4	37	—	—	—	—	—	—	—
		S-Ni Cr 20 3	(Note 2)	3,4	39	—	—	—	—	—	—	—
		S-Ni Cr 20 5 2	(Note 2)	3,4	37	—	—	—	—	—	—	—
		S-Ni 22	(Note 6)	3,4	106	—	—	—	—	—	—	—
		S-Ni Mn 23 4	(Note 6)	3,4	126	—	—	—	—	—	—	—
		S-Ni Cr 30 1	(Note 2)	3,4	37	—	—	—	—	—	—	—
		S-Ni Cr 30 3	(Note 2)	3,4	37	—	—	—	—	—	—	—
		S-Ni 35	(Note 6)	3,4	106	—	—	—	—	—	—	—
		S-Ni Cr 35 3	(Note 2)	3,4	37	—	—	—	—	—	—	—

NOTES:

1 Design strengths are based on separately-cast samples in thicknesses up to 20 mm and cast-on samples for larger thicknesses. Refer to AS 1830 for further details.

2 Pressure limits are 1.1 MPa to 230°C for steam and gases, 1.1 MPa to 190°C for liquids, 1.7 MPa for liquids at less than their boiling point and at a maximum of 50°C.

3 See Clause 3.3.3 for design strength at lower temperatures. See Clause 2.5.3.1 for service limits.

4 To these design strength values a casting quality factor is to be applied.

5 The values relate to an as cast test bar diameter of 30 mm. This corresponds to a relevant wall thickness of 15 mm. Refer to AS 1830 for further details.

6 Pressure limits are 7 MPa for steam, water, oil, air and refrigerants.

7 The diameter of the test piece shall be representative of the relevant wall thickness of the casting.

8 Design strength values are for 12 or 15 mm diameter test bar.

9 Based on mechanical properties measured on test pieces machined from cast-on samples.

TABLE B1(E)

A1

DESIGN TENSILE STRENGTH FOR CLASS 1, 2A, 2B AND 3 VESSELS (MPa)—COPPER AND COPPER ALLOYS

Common name	Product form	Specification	Alloy or UNS No.	Condition or temper	Size or thickness mm	Notes	Design tensile strength, MPa Temperature, °C									
							40	50	100	150	200	250	300	325	350	375
PLATE, SHEET AND STRIP																
Copper–ETP (tough pitch)	Plate, sheet and strip	ASTM B 152	C10200	O25	All	1	46	43	37	34	22	—	—	—	—	—
Copper–OF (oxygen free)	Plate, sheet and strip	ASTM B 152	C11000	O25	≤ 50	1	46	43	37	34	22	—	—	—	—	—
Copper– DHP phosphorus deoxidized	Plate, sheet and strip	ASTM B 152	C12200	O25	All	1	46	43	37	34	22	—	—	—	—	—
Admiralty brass	Plate	ASTM B 171	C44300	O25	≤100	2	69	69	69	69	29	9	—	—	—	—
Naval brass	Plate	ASTM B 171	C46400	O25	≤75	2	92	92	92	91	20		—	—	—	—
Aluminium bronze	Plate	ASTM B 171	C61400	O25	50 < t ≤125	—	128	128	127	125	124	118	—	—	—	—
					≤50	—	138	138	136	134	133	128	—	—	—	—
Aluminium nickel bronze	Plate	ASTM B 171	C63000	O25	89 < t ≤125	4	138	138	135	132	130	124	99	74	55	39
					50 < t ≤89	4	152	151	148	145	143	137	98	74	55	39
					≤50	4	165	165	162	158	156	151	97	74	55	39
90/10 Copper-nickel	Plate	ASTM B 171	C70600	M20	All	—	69	68	65	62	60	—	—	—	—	—
	Plate, sheet				≤64	—	69	68	65	62	60	57	—	—	—	—
	Plate				All	3	69	68	65	62	60	57	45	39	—	—
	Plate	ASTM B 171	C70600	O25	All	—	69	68	65	62	60	—	—	—	—	—
	Plate, sheet				≤64	—	69	68	65	62	60	57	—	—	—	—
	Plate				All	3	69	68	65	62	60	57	45	39	—	—
70/30 Copper-nickel	Plate, sheet	ASTM B 171	C71500	O25	64 < t ≤125	—	83	82	77	74	71	69	67	66	65	65
					≤64	—	92	91	86	83	80	76	74	74	73	72
Silicon bronze	Plate, sheet	ASTM B 96	C65500	O61	≤50	6	83	82	81	68	5	—	—	—	—	—

NOTES:

1 Design strengths for temperatures of 150°C and above are obtained from time dependant properties.

2 Design strengths for temperatures of 175°C and above are obtained from time dependant properties.

3 Design strengths for temperatures of 260°C and above are obtained from time dependant properties.

4 Design strengths for temperatures of 290°C and above are obtained from time dependant properties.

5 Design strengths at intermediate temperatures may be obtained by linear interpolation.

6 Copper-silicon alloys are not always suitable when exposed to certain media and high temperatures, particularly steam above 100°C. The user should ensure that the alloy is satisfactory.

TABLE B1(F)
DESIGN TENSILE STRENGTH FOR CLASS 1, 2A, 2B AND 3 VESSELS (MPa)—ALUMINIUM AND ALUMINIUM ALLOYS

Specification	Class/condition	UNS No.	Thickness mm	Notes	Design tensile strength, MPa						
					Temperature, °C						
					40	50	100	150	200	250	300
ASTM B209 Plate and sheet	T4	Alclad 6061	$1.3 \le t < 6.35$	1, 2, 3	54	54	53	43	30	—	—
	T451	Alclad 6061	$6.35 \le t < 12.7$	1, 2, 3	54	54	53	43	30	—	—
			$12.7 \le t \le 76.2$	2, 3, 4	54	54	53	43	30	—	—
	T4 wld.	Alclad 6061	$1.3 \le t < 6.35$	2, 5	41	41	41	38	25	—	—
	T451 wld.	Alclad 6061	$6.35 \le t < 76.2$	2, 5	41	41	41	38	25	—	—
	T6	Alclad 6061	$1.3 \le t < 6.35$	1, 2, 3	75	75	74	52	29	—	—
	T651	Alclad 6061	$6.35 \le t < 12.7$	1, 2, 3	75	75	74	52	29	—	—
			$12.7 \le t \le 101.6$	2, 3, 4	75	75	74	52	29	—	—
			$101.6 \le t \le 127$	2, 3, 6	71	71	69	51	29	—	—
	T6 wld.	Alclad 6061	$1.3 \le t < 6.35$	2, 5	41	41	41	38	25	—	—
	T651 wld.	Alclad 6061	$6.35 \le t \le 12.7$	2, 5	41	41	41	38	25	—	—
	T4	A96061	$1.3 \le t < 6.35$	2, 3, 7	59	59	58	47	33	—	—
	T451	A96061	$6.35 \le t \le 76.2$	2, 3, 7	59	59	58	47	33	—	—
	T6	A96061	$1.3 \le t < 6.35$	2, 3, 7	83	83	81	58	33	—	—
	T651	A96061	$6.35 \le t \le 101.6$	2, 3, 7	83	83	81	58	33	—	—
			$101.6 < t \le 152.4$	2, 3, 7	79	79	77	56	32	—	—
	T4 wld.	A96061	$1.3 \le t < 6.35$	2, 5, 7	41	41	41	38	25	—	—
	T451 wld.	A96061	$6.35 \le t \le 76.2$	2, 5, 7	41	41	41	38	25	—	—
	T6 wld.	A96061	$1.3 \le t < 6.35$	2, 5, 7	41	41	41	38	25	—	—
	T651	A96061	$6.35 \le t \le 152.4$	2, 5, 7	41	41	41	38	25	—	—

NOTES:

1. Design strengths are 90 percent of those for corresponding core material.
2. For stress relieved tempers (T351 etc) design strength for materials in the basic temper shall be used.
3. Design strength values given are not applicable where welding or thermal cutting is used.
4. The tension test specimen from plate 13 mm and thicker is machined from the core and does not include the cladding alloy; therefore the allowable stress values for thicknesses less than 13 mm shall be used.
5. Reduced section tensile specimen is required to qualify welding procedure.
6. The tension test specimen from plate 13 mm and thicker is machined from the core and does not include the cladding alloys; therefore the allowable stress values shown are 90% of those for the core materials of the same thickness.
7. Design strengths for temperatures of 175°C and above are obtained from time dependant properties.

TABLE B1(G)

DESIGN TENSILE STRENGTH FOR CLASS 1, 2A, 2B AND 3 VESSELS (MPa) NICKEL AND HIGH NICKEL ALLOYS

ASTM Spec No.	Type/ Grade/ UNS No.	Nominal composition	Condition	Size mm	Notes	Design tensile strength, MPa																						
						Temperature, °C																						
						40	50	100	150	200	250	300	325	350	375	400	425	450	475	500	525	550	575	600	625	650	675	700
B 127	N04400	67Ni-30Cu	Annealed	All	—	129	126	112	105	101	101	101	101	101	101	100	99	80	60	42	—	—	—	—	—	—	—	—
			As rolled	All	1	148	148	148	148	148	148	148	148	147	141	127	102	65	35	8	—	—	—	—	—	—	—	—
B 424	N08825	42Ni-21.5Cr-5Mo-2.3Cu	Plate, sheet, strip	All	—	161	158	146	140	134	129	124	122	120	119	119	117	117	117	116	115	113	—	—	—	—	—	—
				All	2	161	161	161	161	161	161	161	161	161	161	160	159	158	157	156	155	153	—	—	—	—	—	—
B 575	N10276	54Ni-16Mo-15Cr	Plate, sheet, strip – Solution annealed	All	2	188	188	188	188	188	187	177	172	169	165	162	159	157	156	155	154	145	118	99	82	67	55	42
				All	—	188	185	170	158	148	139	131	128	125	122	120	118	117	115	115	114	114	112	99	82	67	55	42
B 575	N06022	55Ni-21Cr-13.5Mo	Plate, sheet, strip – Solution annealed	All	2, 3	197	197	197	194	188	183	180	179	178	176	175	174	173	172	171	169	167	148	116	84	66	53	41
				All	3	197	195	182	169	159	150	142	140	137	135	133	131	130	129	128	127	126	124	116	84	66	53	41

NOTES:

1 For plates only.

2 Due to relatively low yield strength of these materials, these higher design strength values were established at temperatures where the short time tensile properties govern to permit the use of these alloys where slightly deformation is acceptable. These higher design strength values exceed 66.7 percent but do not exceed 90 percent of the yield strength at temperature. Use of these design strengths may result in dimensional changes due to permanent strain. These design strength values are not recommended for the flanges of gasketed joints or other applications where slight amounts of distortion can cause leakage or malfunction.

3 This alloy in the solution annealed condition is subjected to severe loss of impact strength at room temperature after exposure in the range of 550°C to 675°C.

TABLE B1(H)

**DESIGN TENSILE STRENGTH FOR CLASS 1, 2A, 2B AND 3 VESSELS (MPa)
TITANIUM AND TITANIUM ALLOYS**

ASTM Spec No.	Type/ Grade/ UNS No.	Nominal composition	Condition	Size mm	Design tensile strength, MPa Temperature, °C							
					40	100	150	200	250	300	325	350
PLATE, SHEET AND STRIP												
B 265	1	Ti	Annealed	All	69	67	56	45	38	34	27	23
	2	Ti	Annealed	All	99	97	84	71	61	54	47	44
	3	Ti	Annealed	All	128	125	106	88	72	60	53	50
	11	Ti-Pd	Annealed	All	69	67	56	45	38	34	27	23
	17	Ti-Pd	Annealed	All	69	67	56	45	38	34	27	23
	27	Ti-Ru	Annealed	All	69	67	56	45	38	34	27	23
	7	Ti-Pd	Annealed	All	99	97	84	71	61	54	47	44
	16	Ti-Pd	Annealed	All	99	97	84	71	61	54	47	44
	26	Ti-Ru	Annealed	All	99	97	84	71	61	54	47	44
	12	Ti-0.3Mo-0.8Ni	Annealed	All	138	138	126	111	100	91	86	84
	9	Ti-3Al-2.5V	Annealed	All	177	177	168	155	141	127	120	118

TABLE B2

DESIGN TENSILE STRENGTH (MPa) FOR FLANGE BOLTING

Material							Design tensile strength, MPa																			
							Temperature, °C																			
Type	Spec	Grade	UNS	Class	Dia, mm	T_R, °C	50	100	150	200	250	300	325	350	375	400	425	450	475	500	525	550	575	600	625	650
C, C-Mn steel	AS 1110	4.6	—	—	All	−29	60	60	60	60	50	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		5.6			All		75	75	75	75	72	65	—	—	—	—	—	—	—	—	—	—	—	—	—	—
C, C-Mn steel	AS 4291.1	4.6	—	—	All	−29	60	60	60	60	50	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		5.6			All		75	75	75	75	72	65	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		8.8		Q&T	All		160	160	160	160	160	160	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		10.9		Q&T	All		200	200	200	200	200	200	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		12.9		Q&T	All		240	240	240	240	240	240	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Carbon steel	ASTM A 325	—	K02706	—	—	−29	140	140	140	140	133	127	125	122	—	—	—	—	—	—	—	—	—	—	—	—
		1	K02706	—	$12 < d < 26$		159	159	159	159	151	145	142	139	—	—	—	—	—	—	—	—	—	—	—	—
					$32 < d < 39$		140	140	140	140	133	127	125	122	—	—	—	—	—	—	—	—	—	—	—	—
	ASTM A 354	BC	K04100	—	$6 < d < 64$	−18	172	172	172	172	172	172	172	172	—	—	—	—	—	—	—	—	—	—	—	—
					$65 < d < 102$		159	159	159	159	159	159	159	159	—	—	—	—	—	—	—	—	—	—	—	—
	ASTM A 354	BD	K04100	—	$6 < d < 65$	−7	207	207	207	207	207	207	207	207	—	—	—	—	—	—	—	—	—	—	—	—
					$65 < d < 102$		193	193	193	193	193	193	193	193	—	—	—	—	—	—	—	—	—	—	—	—
	ASTM A 449	—	K04200	—	$d < 26$	−29	159	159	159	159	151	145	142	139	136	—	—	—	—	—	—	—	—	—	—	—
					$25 < d < 39$		140	140	140	140	133	127	125	122	120	—	—	—	—	—	—	—	—	—	—	—
					$40 < d < 77$		100	100	100	100	100	100	100	98	95	—	—	—	—	—	—	—	—	—	—	—
C-0.25Mo	ASTM A 320	L7A	G40370	—	$d < 65$	Note 2	172	172	172	172	172	172	172	172	—	—	—	—	—	—	—	—	—	—	—	—

(*continued*)

TABLE B2 (*continued*)

Material							Design tensile strength, MPa																			
							Temperature, °C																			
Type	**Spec**	**Grade**	**UNS**	**Class**	**Dia, mm**	**T_R, °C**	**50**	**100**	**150**	**200**	**250**	**300**	**325**	**350**	**375**	**400**	**425**	**450**	**475**	**500**	**525**	**550**	**575**	**600**	**625**	**650**
C-0.25Mo	ASTM A 574	4037	G40370	—	$d<13$	—	241	241	241	241	241	241	—	—	—	—	—	—	—	—	—	—	—	—	—	—
					$d>15$		233	233	233	233	233	233	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		4042	G40420	—	$d<13$	—	241	241	241	241	241	241	—	—	—	—	—	—	—	—	—	—	—	—	—	—
					$d>15$		233	233	233	233	233	233	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		4140	G41400	—	$d>15$	—	233	233	233	233	233	233	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1Cr-0.2Mo	ASTM A 193	B7	G41400	—	$d<65$	−48	172	172	172	172	172	172	172	172	172	162	146	118	93	69	44	19	—	—	—	—
					$65<d<102$	−40	159	159	159	159	159	159	159	159	159	153	139	117	93	69	44	19	—	—	—	—
					$102<d<180$		129	129	129	129	129	129	129	129	129	127	122	115	94	68	44	19	—	—	—	—
		B7M	G41400	—	$d<65$	−48	138	138	138	138	138	138	138	138	138	136	130	123	94	68	44	19	—	—	—	—
1Cr-0.2Mo	ASTM A 320	L7	G41400	—	$d<65$	Note 2	172	172	172	172	172	172	172	172	172	162	146	118	—	—	—	—	—	—	—	—
		L7M	G41400	—	$d<65$	Note 2	138	138	138	138	138	138	138	138	138	136	128	115	94	68	44	19	—	—	—	—
1Cr-0.5Mo-V	ASTM A 193	B16	K14072	—	$d<65$	−29	172	172	172	172	172	172	172	172	172	172	172	164	148	122	92	61	35	14	—	—
					$65<d<102$		152	152	152	152	152	152	152	152	152	152	152	147	133	114	90	61	35	14	—	—
					$102<d<180$		138	138	138	138	138	138	138	138	138	138	138	132	119	105	88	62	34	14	—	—
1Cr-0.5Mo-V	ASTM A 540	B21	K14073	1	$d<102$	T_{CV}	228	228	228	228	228	228	228	228	228	—	—	—	—	—	—	—	—	—	—	—
				2	$d<102$		214	214	214	214	214	214	214	214	214	—	—	—	—	—	—	—	—	—	—	—
				3	$d<153$		200	200	200	200	200	200	200	200	200	—	—	—	—	—	—	—	—	—	—	—
				4	$d<153$		186	186	186	186	186	186	186	186	186	—	—	—	—	—	—	—	—	—	—	—
				5	$d<51$		166	166	166	166	166	166	166	166	166	—	—	—	—	—	—	—	—	—	—	—
					$51<d<204$		159	159	159	159	159	159	159	159	159	—	—	—	—	—	—	—	—	—	—	—
1.75Ni-0.75Cr-0.25Mo	ASTM A 574	4340	G43400	—	$d<15$	—	241	241	241	241	241	241	—	—	—	—	—	—	—	—	—	—	—	—	—	—
					$d>15$		233	233	233	233	233	233	—	—	—	—	—	—	—	—	—	—	—	—	—	—

(*continued*)

TABLE B2 (*continued*)

Material							Design tensile strength, MPa																			
							Temperature, °C																			
Type	Spec	Grade	UNS	Class	Dia, mm	T_R, °C	50	100	150	200	250	300	325	350	375	400	425	450	475	500	525	550	575	600	625	650
2Ni-0.75Cr-0.25Mo	ASTM A 540	B23	H43400	1	$d < 204$	T_{CV}	228	228	228	228	228	228	228	228	228	—	—	—	—	—	—	—	—	—	—	—
				2	$d < 245$	T_{CV}	214	214	214	214	214	214	214	214	214	—	—	—	—	—	—	—	—	—	—	—
				3	$d < 245$	Note 3	200	200	200	200	200	200	200	200	200	—	—	—	—	—	—	—	—	—	—	—
				4	$d < 245$	Note 3	186	186	186	186	186	186	186	186	186	—	—	—	—	—	—	—	—	—	—	—
				5	$d < 153$	Note 4	166	166	166	166	166	166	166	166	166	—	—	—	—	—	—	—	—	—	—	—
					$153 < d < 245$		159	159	159	159	159	159	159	159	159	—	—	—	—	—	—	—	—	—	—	—
2Ni-0.75Cr-0.33Mo	ASTM A 540	B24	K24064	1	All	Note 3	228	228	228	228	228	228	228	228	228	—	—	—	—	—	—	—	—	—	—	—
				2	All	Note 5	214	214	214	214	214	214	214	214	214	—	—	—	—	—	—	—	—	—	—	—
				3	All	Note 6	200	200	200	200	200	200	200	200	200	—	—	—	—	—	—	—	—	—	—	—
				4	All		186	186	186	186	186	186	186	186	186	—	—	—	—	—	—	—	—	—	—	—
				5/115	All	Note 7	159	159	159	159	159	159	159	159	159	—	—	—	—	—	—	—	—	—	—	—
				5/120	All		166	166	166	166	166	166	166	166	166	—	—	—	—	—	—	—	—	—	—	—
5Cr-0.5Mo	ASTM A 193	B5	K50100	—	$d < 102$	−29	138	138	138	138	138	138	138	138	138	138	129	105	78	59	44	34	26	20	13	9
13Cr	ASTM A 193	B6	S41000	—	$d < 102$	−29	152	152	152	152	152	152	152	152	152	152	136	112	89	67	—	—	—	—	—	—
16Cr-12Ni-2Mo	ASTM A 193	B8M	S31600	1	All	−254	52	52	52	50	47	44	43	42	42	41	41	40	40	39	39	39	—	—	—	—
				2	$d < 20$	—	152	152	152	152	143	136	133	130	128	125	123	—	—	—	—	—	—	—	—	—
					$20 < d < 26$		138	138	138	138	128	121	117	113	111	110	108	—	—	—	—	—	—	—	—	—
					$26 < d < 32$		112	112	112	112	112	112	112	112	112	112	112	112	112	110	108	107	—	—	—	—
					$32 < d < 39$		86	86	86	86	86	86	86	86	86	86	86	86	—	—	—	—	—	—	—	—
16Cr-12Ni-2Mo	ASTM A 193	B8M2	S31600	—	$d < 51$	T_{CV}	129	129	129	129	129	129	129	129	129	129	129	129	129	127	125	123	—	—	—	—
					$51 < d < 65$		112	112	112	112	112	112	112	112	112	112	112	112	112	110	108	107	—	—	—	—

(*continued*)

TABLE B2 (*continued*)

Material							Design tensile strength, MPa																			
							Temperature, °C																			
Type	**Spec**	**Grade**	**UNS**	**Class**	**Dia, mm**	**T_R, °C**	**50**	**100**	**150**	**200**	**250**	**300**	**325**	**350**	**375**	**400**	**425**	**450**	**475**	**500**	**525**	**550**	**575**	**600**	**625**	**650**
16Cr-12Ni-2Mo	ASTM A 320	B8M	S31600	1	All	Note 2	52	52	52	50	47	44	—	—	—	—	—	—	—	—	—	—	—	—	—	—
				2	d <20	Note 2	152	152	152	152	143	136	—	—	—	—	—	—	—	—	—	—	—	—	—	—
					20 < d <26		138	138	138	138	128	121	—	—	—	—	—	—	—	—	—	—	—	—	—	—
					26 < d <32		112	112	112	112	112	112	—	—	—	—	—	—	—	—	—	—	—	—	—	—
					32 < d <39		86	86	86	86	86	86	—	—	—	—	—	—	—	—	—	—	—	—	—	—
16Cr-12Ni-2Mo	ASTM A 320	B8MA	S31600	1A	All	Note 2	52	52	52	50	47	44	—	—	—	—	—	—	—	—	—	—	—	—	—	—
18Cr-8Ni	ASTM A 193	B8	S30400	1	All	−254	52	52	51	48	45	43	42	41	40	40	39	38	37	37	36	35	—	—	—	—
				2	d <19	T_{CV}	172	172	172	172	172	172	172	172	172	172	172	172	172	172	169	164	—	—	—	—
					19 < d <25		138	138	138	138	138	138	138	138	138	138	138	138	138	138	138	138	—	—	—	—
					25 < d <32		130	115	112	112	112	112	112	112	112	112	112	112	112	112	112	112	—	—	—	—
					32 < d <38		130	114	103	96	90	87	86	86	86	86	86	86	86	86	86	86	—	—	—	—
18Cr-8Ni	ASTM A 320	B8	S30400	1	All	Note 2	52	52	51	48	45	43	—	—	—	—	—	—	—	—	—	—	—	—	—	—
				2	d <19	Note 2	172	172	172	172	172	172	172	172	172	172	172	172	172	172	169	164	—	—	—	—
					19 < d <25		138	138	138	138	138	138	138	138	138	138	138	138	138	138	138	138	—	—	—	—
					25 < d <32		130	115	112	112	112	112	112	112	112	112	112	112	112	112	112	112	—	—	—	—
					32 < d <38		130	114	103	96	90	87	86	86	86	86	86	86	86	86	86	86	—	—	—	—
		B8A		1A	All	Note 2	52	52	51	48	45	43	—	—	—	—	—	—	—	—	—	—	—	—	—	—
18Cr-8Ni-N	ASTM A 193	B8NA	S30451	1A	All	−196	52	52	51	48	45	43	42	41	40	40	39	38	37	37	36	35	—	—	—	—
18Cr-8Ni-S	ASTM A 320	B8F	S30323	1	All	Note 2	52	52	51	48	45	43	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		B8FA		1A	All	Note 2	52	52	51	48	45	43	—	—	—	—	—	—	—	—	—	—	—	—	—	—
18Cr-10Ni-Cb	ASTM A 193	B8C	S34700	1	All	−254	52	52	52	52	52	50	49	48	48	47	47	47	46	46	46	46	—	—	—	—

(*continued*)

TABLE B2 (*continued*)

Material							Design tensile strength, MPa																			
							Temperature, °C																			
Type	**Spec**	**Grade**	**UNS**	**Class**	**Dia, mm**	**T_R, °C**	**50**	**100**	**150**	**200**	**250**	**300**	**325**	**350**	**375**	**400**	**425**	**450**	**475**	**500**	**525**	**550**	**575**	**600**	**625**	**650**
18Cr-10Ni-Cb	ASTM A 320	B8C	S34700	1	All	Note 2	52	52	52	52	52	50	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		B8CA		1A	All	Note 2	52	52	52	52	52	50	—	—	—	—	—	—	—	—	—	—	—	—	—	—
18Cr-10Ni-Ti	ASTM A 193	B8T	S32100	1	All	–254	52	52	52	52	50	47	46	45	45	44	43	43	42	42	42	41	—	—	—	—
18Cr-10Ni-Ti	ASTM A 320	B8T	S32100	1	All	Note 2	52	52	52	52	50	47	—	—	—	—	—	—	—	—	—	—	—	—	—	—
		B8TA		1A	All	Note 2	52	52	52	52	50	47	—	—	—	—	—	—	—	—	—	—	—	—	—	—
19Cr-9Ni-Mo-W	ASTM A 453	651	S63198	A	d <80	T_{CV}	121	109	102	97	93	89	87	85	83	81	80	79	—	—	—	—	—	—	—	—
					d >80		103	94	87	83	80	76	74	73	71	70	68	67	—	—	—	—	—	—	—	—
				B	d <80	T_{CV}	103	94	87	83	80	76	74	73	71	70	68	67	—	—	—	—	—	—	—	—
					d >80		86	78	73	69	66	63	62	60	59	58	57	57	—	—	—	—	—	—	—	—
25Ni-15Cr-2Ti	ASTM A 453	660	S66286	A	All	T_{CV}	147	147	147	147	147	147	147	147	147	147	147	147	—	—	—	—	—	—	—	—
				B	All		147	147	147	147	147	147	147	147	147	147	147	147	—	—	—	—	—	—	—	—

A1 NOTES:

1 Stresses at intermediate temperatures may be obtained by linear interpolation.

2 $T_R = T_{CV} - 3$°C.

3 For $d \leq 152$ mm, $T_R = T_{CV} - 3$°C. For $d > 152$ mm, $T_R = T_{CV}$.

4 For $d \leq 203$ mm, $T_R = T_{CV} - 3$°C. For $d > 203$ mm, $T_R = T_{CV}$.

5 For $d \leq 178$ mm, $T_R = T_{CV} - 3$°C. For $d > 178$ mm, $T_R = T_{CV}$.

6 For $d \leq 205$ mm, $T_R = T_{CV} - 3$°C. For $d > 205$ mm, $T_R = T_{CV}$.

7 For $d \leq 241$ mm, $T_R = T_{CV} - 3$°C.

TABLE B3

YOUNG'S MODULUS (MODULUS OF ELASTICITY) (*E*)

Material		Young's modulus, GPa														
		Temperature, °C														
Type or grade	**Nominal composition**	**–200**	**–150**	**–100**	**–50**	**0**	**50**	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**450**	**500**
Carbon and low alloy steels	C ≤ .3%C	217	213	210	207	204	201	198	195	192	189	186	179	171	162	150
	C > .3%C	215	212	209	206	203	200	197	194	191	187	184	178	170	161	149
	C-.5Mo, Mn-.5Mo, Mn-.25Mo, Mn-V	215	211	208	205	202	199	196	193	190	187	184	178	170	160	150
	.5Ni-.5Mo-V, .5Ni-.5Cr-.25Mo-V, .75Ni-.5Mo-Cr-V, .75Ni-1Mo-.75Cr, .75Ni-.5Cu-Mo, 1Ni-.4Cr-.5Mo, .75Cr-.5Ni-Cu, .75Cr-.75Ni-Cu-Al, 2Ni-1Cu, 2.5Ni, 3.5Ni.	204	201	198	196	193	190	187	184	181	178	175	171	167	163	159
	.5Cr-.5Mo, 1Cr-.5Mo, 1.25Cr-.5Mo(+Si), 2Cr-.5Mo	218	215	212	210	207	204	200	196	193	190	187	183	179	174	170
	2.25Cr-1Mo, 3Cr-1Mo	225	222	218	215	212	209	206	203	199	196	192	188	184	179	175
	5Cr-.5Mo(+Si, +Ti), 7Cr-.5Mo, 9Cr-Mo	—	—	—	219	215	211	207	204	201	198	194	190	190	176	168
Stainless steels																
405, 410, 429, 430	12Cr-Al, 13Cr, 15Cr, 17Cr	215	213	210	206	202	199	196	192	189	185	181	178	174	166	156
304 316, 317 321 347 and 348 309, 310	18Cr-8Ni 16Cr-12Ni-2Mo, 18Cr-13Ni-3Mo 18Cr-10Ni-Ti 18Cr-10Ni-Nb 23Cr-12Ni, 25Cr-12Ni,25Cr-20Ni	209	206	203	200	197	194	190	186	183	179	175	172	169	164	161
S31803, 2304	22Cr-.5Ni-.3Mo, 23Cr-4Ni	—	—	—	205	200	195	190	185	180	175	170	165	160	—	—
N08904	25Ni-20Cr-4.5Mo-1.5Cu	—	—	—	200	196	193	189	185	181	167	172	168	165	—	—
N08028	31Ni-27Cr-3.5Mo-1.0Cu	—	—	—	204	201	198	195	192	189	185	180	175	170	—	—
Aluminium alloys																
3003, 3004, 6061, 6063		77	75	73	72	70	68	66	63	60	—	—	—	—	—	—
5052, 5054		78	76	75	73	71	69	67	65	62	—	—	—	—	—	—
5083, 5086		79	77	76	74	72	70	68	65	62	—	—	—	—	—	—

(continued)

TABLE B3 (*continued*)

Material			Young's modulus, GPa														
			Temperature, °C														
Type or grade	**Nominal composition**		**–200**	**–150**	**–100**	**–50**	**0**	**50**	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**450**	**500**
Copper and copper alloys																	
C21000	Copper >95%		124	123	121	117	114	111	108	105	102	99	95	2	89	86	83
C22000, C24000	Brasses:	10 and 20Zn	124	123	121	120	118	116	114	111	109	106	103	101	98	97	94
C26000, C28000		30 and 40Zn	110	108	107	106	104	101	99	97	95	93	91	89	86	83	80
C70610	Cu-Ni:	10Ni	131	130	128	126	125	123	121	119	116	112	110	107	104	101	98
C71630		20 and 30Ni	161	159	157	152	148	144	140	137	133	129	124	120	116	112	108
C64250	Bronze		116	114	112	110	107	104	102	99	96	93	89	86	84	81	78
Nickel and nickel alloys																	
200, 201	Ni and Low C Ni		222	218	215	211	208	205	202	199	197	194	192	189	186	182	179
330	Ni-44Fe-18Cr-1Si		207	204	201	197	194	191	188	185	183	181	179	177	174	170	167
400 and 405	Ni-32Cu		192	189	186	184	181	178	175	173	171	168	166	164	161	158	155
600	Ni-15.5Cr-8Fe		229	226	222	219	215	211	208	206	204	201	199	196	192	189	185
800 and 800H	Ni-46Fe-21Cr		210	207	204	200	197	194	191	189	187	185	183	180	177	174	170
825	Ni-30Fe-21Cr-3Mo-2Cu		207	204	201	197	194	191	188	185	183	181	179	177	174	170	167
B	Ni-28Mo-5Fe		230	226	223	219	215	212	209	206	204	201	199	197	193	189	185
C-4	Ni-16Cr-16Mo		220	217	214	209	206	203	200	197	195	193	191	188	185	181	177
C276	Ni-15.5Cr-16Mo-5.5Fe-4W		220	217	214	209	206	203	200	197	195	193	191	188	185	181	177
Titanium and titanium alloys																	
1, 2, 3 and 7			—	—	—	110	108	106	103	100	97	93	88	84	80	—	—
Zirconium and zirconium alloy 702	Zr		—	—	—	101	100	98	95	92	86	80	74	68	—	—	—
705 and 706	Zr-2.5Nb		—	—	—	103	102	100	93	86	80	75	71	67	—	—	—

NOTES:

1 These values are recommended for calculation purposes. It is not implied that materials are suitable for all temperatures shown.

2 Data based on ASME BPV III-1, with additions from the *ASM Handbook*.

3 Values at intermediate temperatures may be obtained by linear interpolation.

4 Values at temperatures beyond those listed are to be used by agreement between parties concerned.

TABLE B4

LINEAR THERMAL EXPANSION

Material type	Mean coefficients of Linear Thermal Expansion for metals from 20°C to the temperature indicated, mm/mm/°C × 10^{-6}																											
	Temperatures, °C																											
	−125	−100	−50	0	50	100	150	200	250	300	325	250	375	400	425	450	475	500	525	550	575	600	625	650	675	700	725	750
Plain Carbon steel, C-Mn steel	10.1	10.4	10.8	11.3	11.8	12.1	12.4	12.7	13.0	13.3	13.4	13.5	13.7	13.8	14.0	14.1	14.2											
C-Si steel, C-0.5Mo, 1Cr-0.5Mo	10.1	10.1	10.2	10.3	10.5	11.0	11.6	12.1	12.6	13.0	13.2	13.4	13.5	13.7	13.9	14.0	14.1	14.2	14.3									
C-Mn-Si steel, 1.25Cr-0.5Mo, 3Cr-Mo					10.1	10.7	11.3	11.9	12.3	12.8	13.0	13.2	13.4	13.5	13.7	13.8	13.9	14.1	14.2	14.3	14.5	14.6	14.7					
Mn-Mo steel	10.1	10.5	11.3	12.1	12.8	13.1	13.4	13.6	13.8	14.0	14.1	14.2	14.3	14.4	14.5	14.6	14.6	14.7	14.8	11.6	4.9							
2.5Ni, 3.5Ni					11.4	11.8	12.2	12.5	12.8	13.1	13.2	13.3	13.5	13.6	13.7													
2.25Cr-Mo	10.1	10.4	10.8	11.3	11.8	12.1	12.4	12.7	13.0	13.2	13.3	13.4	13.5	13.6	13.7	13.8	13.9	14.0	14.0	14.1	14.2	14.2	14.3					
5Cr-0.5Mo	10.1	10.4	10.8	11.3	11.8	12.1	12.4	12.5	12.7	12.8	12.9	13.0	13.0	13.1	13.2	13.2	13.3	13.4	13.4	13.5	13.6	13.6	13.7					
7Cr-0.5Mo, 9Cr-Mo	10.1	10.2	10.3	10.4	10.6	10.9	11.1	11.3	11.5	11.7	11.8	11.8	11.9	12.0	12.1	12.2	12.3	12.3	12.4	12.5	12.5	12.6	12.7					
12Cr, 13Cr	9.2	9.5	9.9	10.4	10.8	11.1	11.3	11.5	11.6	11.7	11.8	11.8	11.9	11.9	12.0	12.0	12.1	12.1	12.2	12.2	12.3	12.3	12.3					
15Cr, 17Cr	9.2	9.3	9.4	9.6	9.7	10.0	10.2	10.3	10.5	10.7	10.7	10.8	10.9	11.0	11.0	11.1	11.2	11.2	11.3	11.4	11.4	11.5	11.5					
17-19Cr (TP 439)						10.1	10.3	10.4	10.6	10.8	10.9	11.0	11.1	11.2	11.2	11.3	11.4	11.5	11.6	11.7	11.8	11.8	11.9	12.0	12.0	12.1	12.2	12.3
All grades of TP 316 and 317 Stainless steel					15.5	15.8	16.2	16.5	16.9	17.2	17.3	17.5	17.6	17.7	17.8	17.9	18.0	18.1	18.2	18.3	18.4	18.5	18.6	18.7	18.8	18.9	19.1	19.1
All grades of TP 304 Stainless steel					15.5	15.9	16.2	16.5	16.8	17.1	17.2	17.3	17.5	17.6	17.7	17.8	17.9	18.0	18.1	18.2	18.3	18.3	18.4	18.5	18.6	18.7	18.8	18.8
All grades of TP 321 Stainless steel					16.3	16.5	16.7	16.8	16.9	17.0	17.1	17.1	17.2	17.2	17.3	17.3	17.4	17.4	17.5	17.5	17.6	17.6	17.7	17.7	17.8	17.8	17.9	17.9
All grades of TP 347 Stainless steel					15.6	16.1	16.6	17.0	17.3	17.6	17.7	17.9	18.0	18.0	18.1	18.2	18.4	18.5	18.5	18.6	18.7	18.8	18.9	19.0	19.1	19.2	19.3	19.3
25Cr-12Ni, 23Cr-12Ni, 25Cr-20Ni					16.0	16.3	16.4	16.4	16.5	16.6	16.6	16.6	16.7	16.7	16.7	16.7	16.8	16.8	16.8	16.9	16.9	17.0	17.0	17.1	17.1	17.2	17.3	17.3
AL-6XN (N08367)						15.3	15.4	15.5	15.1	15.6	15.9	15.9	16.0	16.0	16.1	16.2	16.3	16.4	16.5	16.6	16.7	16.7	16.8	16.9	17.0	17.1	17.3	17.4
Aluminium (3003)	21.3	21.5	21.9	22.3	22.7	23.2	23.7	24.2																				
Aluminium (6061)	21.3	21.5	21.9	22.3	22.8	23.3	23.8	24.3																				
Titanium (Grade 1,2,3 and 7)					8.4	8.5	8.6	8.6	8.7	8.8	8.8	8.9	9.0	9.0	9.1													
Ni-Cu (N04400)					14.1	14.6	15.0	15.3	15.6	15.8	15.9	15.9	16.0	16.0	16.0													
Ni-Cr-Fe (N06600)					12.5	13.0	13.3	13.6	13.8	14.0	14.1	14.2	14.3	14.4	14.5													
Ni-Fe-Cr (N08800 and N08810)					14.5	15.1	15.5	15.8	16.0	16.2	16.2	16.3	16.4	16.5	16.6	16.6	16.7	16.8	16.9									

(continued)

TABLE B4 (*continued*)

Material type	Mean coefficients of Linear Thermal Expansion for metals from 20°C to the temperature indicated, mm/mm/°C × 10^{-6}																											
	Temperatures, °C																											
	–125	**–100**	**–50**	**0**	**50**	**100**	**150**	**200**	**250**	**300**	**325**	**250**	**375**	**400**	**425**	**450**	**475**	**500**	**525**	**550**	**575**	**600**	**625**	**650**	**675**	**700**	**725**	**750**
Ni-Fe-Cr-Mo-Cu (N08825)					11.0	3.5	14.1	14.3	14.5	14.7	14.8	14.9	15.0	15.0	15.1													
Ni-Mo (Alloy B)					11.0	11.3	11.4	11.5	11.5	11.6	11.7	11.8	11.8	11.9	12.0													
Ni-Mo-Cr (Alloy C-276) (N10276)					11.0	11.4	11.7	12.0	12.4	12.7	12.8	12.9	13.0	13.1	13.2													
Nickel (Alloy 200) (N02200)	11.2	11.3	11.6	12.0	12.4	13.0	13.5	13.9	14.2	14.4	14.5	14.6	14.7	14.8	14.9	15.1	15.2	15.4	15.4	15.5	15.6	15.7	15.8	15.8	15.9	16.0	16.0	16.0
SAF2205 (S31803)					12.6	12.7	13.1	13.5	13.7	14.0	14.1	14.2	14.2	14.3	14.4													
3RE60 (S31500)					14.5	14.9	15.2	15.5	15.7	15.9	16.1	16.2	16.3	16.4	16.5													
70-30 Cu-Ni (C71500)						15.3	15.7	16.0	16.3																			
90-10 and 80-20 Cu-Ni							17.1																					
Copper	15.5	15.9	16.4	16.7	17.0	17.3	17.5	17.6	17.8	18.1	18.2	18.3	18.4	18.5	18.5	18.6	18.7	18.8	18.9									
Brass	16.4	16.6	16.9	17.1	17.3	17.5	18.0	18.3	18.8	19.2	19.3	19.5	19.7	19.9	20.1	20.3	20.5	20.6	20.8	21.0	21.2	21.5	21.6					
Aluminium Bronze									16.2																			
7Mo (S32900)						10.2	10.8	11.0	11.1	11.4	11.5	11.6	11.7	11.9	12.0	12.2	12.3	12.5	12.6	12.8	12.9	13.1	13.2	13.4	13.5	13.7	13.9	14.0
7Mo Plus (S32950)									11.9	12.4	12.6	12.8	13.0	13.2	13.5	13.6	13.8	13.9	14.1	14.2	14.3	14.4	14.5	14.5	14.6	14.6	14.7	14.7
Copper-Silicon									18.0																			
Admiralty										20.2																		
Zirconium					5.8	6.1	6.3	6.6	7.0	7.1	7.2	7.3																
Cr-Ni-Fe-Mo-Cu-Cb (Alloy 20CB)					14.9	15.0	15.2	15.5	15.7	16.0	16.1	16.2	16.4	16.5	16.6	16.8	16.9	16.9	17.0	17.0	17.1	17.1	17.2	17.2	17.2	17.3		
Ni-Cr-Mo-Cb (Alloy 625) (N06625)	9.5	10.3	11.3	11.8	12.2	12.8	13.0	13.1	13.2	13.4	13.4	13.5	13.5	13.6	13.7	13.8	13.8	13.9	14.0	14.1	14.3	14.4	14.6	14.8	14.9	15.0	15.2	15.3
Al 29-4-2						9.4																						
Sea-Cure						9.7	9.8	10.1	10.4	10.5	10.6	10.7																

Data Source is TEMA Table D-11 M.

APPENDIX C

RISK MANAGEMENT

(Informative)

C1 INTRODUCTION

Australian Government regulations generally require the designer and/or the manufacturer to identify hazards, assess risks, consider risk reduction and control risk of hazardous plant and equipment. Specific requirements may vary between the states and territories.

For the purpose of this Standard it is intended the 'designer/manufacturer' of a pressure vessel includes the following:

(a) 'Plant designer', working on behalf of the purchaser, owner, project developer (or similar), who determines items including the overall plant design, products, process, capacity, location, and life.

(b) 'Process designer', who determines items including the vessel pressure, temperature, size, shape, and flow.

(c) 'Piping and instrument designer', who determines items including the size, number and position of nozzles.

(d) 'Vessel designer', who determines the mechanical design of pressure parts and appurtenances to comply with the purchasers' requirements (via the design specifications) and with this Standard.

(e) 'Vessel fabricator', who fabricates or manufactures according to the design.

The term designer/manufacturer may cover a number of separate parties, a combination of parties, or be a single person or body, who determines all requirements and constructs and supplies the equipment.

The parties above should jointly ensure suitable risk management (for the risks they can control) and contribute data to enable the manufacturer to complete Form C1 in relation to the vessel.

Most of the risk assessment work should have already been done by the plant designer prior to issuing the specification for the construction, design and manufacture of the vessel. This work would include compliance with the requirements of this Standard which—

(i) automatically requires consideration of almost all hazards, modes of failure and service conditions for which the designer is directly concerned;

(ii) has inherently analysed and assessed risks and the normal risks associated with the above hazards; and

(iii) has provided suitable controls for these risks comparable with current world good practice.

This Appendix gives guidance for the designer/manufacturer on how to comply with these requirements. It takes into account the relevant requirements in AS 3873, AS/NZS ISO 31000 and the WorkSafe guide, *Plant design—Making it safe*.

C2 DESIGNER'S ROLE

The designer(s) in the context of this Standard is responsible for the mechanical design of pressure parts and appurtenances for conformance with this Standard and the design or vessel specification. Selection of pressure, temperature, size, flow, and controls is normally the responsibility of the owner/process designer while selection of materials, location, installation, security and the like, may be the responsibility of the plant designer.

The designer with respect to AS 1210 should—

(a) recognize that mechanical design is an important part of the overall control of risk associated with the vessel; and

(b) at the design stage, consider all hazards which may feasibly arise with the vessel during its entire life and for which the designer of pressure parts can control or influence.

C3 RISK MANAGEMENT SYSTEM

C3.1 System

The designer should have a documented risk management system that is applicable to the vessels designed.

NOTE: AS 3873, Appendix D provides some guidance.

Figure C1 gives a simple form to cover risk management with mechanical design to AS 1210 for most pressure vessels. This form may be used to document the overall risk assessment.

NOTE: To describe the 'likelihood' qualitative descriptions such as 'rare', 'unlikely', 'likely'or 'almost certain' are appropriate.

C3.2 Hazards

The designer should list any special hazards not already addressed adequately in design to this Standard and related Standards, e.g. bursting, leakage and distortion. (See Clause 3.1.4 and AS/NZS 3788).

The hazard level in accordance with AS 4343 should be determined and recorded. Where the contents or location are not known, assumed contents and location should be specifically stated.

C3.3 Assessment

Initial hazard analysis and risk assessment should be carried out at the equipment specification stage by the plant designer or owner. The assessment of risk is largely done in the design (e.g. loads and safety factors) to this Standard (AS 1210) but special factors should be listed e.g. if the vessel will be located near large numbers of people or in a hazardous location, the contents are lethal, abnormal additional loads will be imposed or if suitable personal protective equipment may be needed.

C3.4 Control

Risk controls should be implemented in the specification and design of the equipment constructed to this Standard (AS 1210) and any special controls should be listed. These should include any special control measures that may deal with lifting, transport, installation, operation, inspection (e.g. as the creep or fatigue life is approached), maintenance and the like.

C3.5 Record

The final record of the assessment should be signed, dated and verified in a similar manner to that used with drawings and calculations.

In Figure C1, 'Section C4' of the assessment form is intended to indicate the vessel has been designed on the basis that reasonable care will be taken of the vessel during its operating life (see Clause 1.8.6), which may include compliance with appropriate Standards as listed on the form.

C3.6 Supply of information

Consideration should be given to the supply of the completed form (see Paragraph C3.1) by the designer or owner to the manufacturer as a means of assuring the purchaser and of assisting to cover the risk management required by the owner/user.

See also Appendix F for information to be supplied by the designer.

C4 MANUFACTURER'S ROLE

The manufacturer should:

(a) Record any hazard, risk management, or risk control (additional to that in Paragraph C3) considered by the manufacturer to be necessary to ensure vessels are safe.

(b) Report any design faults that may affect safety to the designer for corrective action.

(c) Comply with Appendix F for information to be supplied by the manufacturer.

GENERIC RISK ASSESSMENT FORM FOR PRESSURE VESSELS

C1 EQUIPMENT IDENTIFICATION Registered design No. ....Registered No. ....Plant/Equipment No. ....

Workplace/Facility .... Vessel Title .... Date Built: ....

C2 HAZARD IDENTIFICATION (i.e. possible sources of harm)

(a) Pressure energy (contributing to leakage and rupture) Hazard Level. ....

(b) Escape of flammable, toxic or harmful fluid from vessel, components and joints—with risk of subsequent fire, burns, injury or harm to health, property or environment. ....

(c) Rapid release of high pressure fluid—with projection of vessel components or fragments or high pressure blast. ....

(d) Collapse, vacuum implosion or detachment of vessel from equipment or vehicle (in operation or accident)—with projection of vessel and leakage. ....

(e) Furnace explosion with fired vessels—with blast, projection of flame or parts. ....

(f) Domino or knock-on effect—with subsequent damage to adjacent plant or property. ....

(g) Associated hazards—machinery, collision, fire, fall, confined space etc. ....

(h) Special operational hazards e.g. quick actuating door, vehicle, or site. ....

(i) Other (owner-user to add if applicable). ....

C3 RISK ASSESSMENT (i.e. likelihood or chance of above events and consequences)

Likelihood

(a) Vessel (protective devices and controls)—chance of failure and above event(s) is: ....

(b) Installation—chance of failure and above event(s) is: ....

(c) Operation, care and maintenance—chance of failure and above event(s) but depends on owner-user, is: ....

(d) Periodic inspection—chance of undetected deterioration is: ....

(e) Emergency action—chance of inadequate action is: ....

Consequences

(f) Persons possibly effected—operator(s) and others in vicinity. ....

(g) Possible effects—blast injury, burns, or death to persons, loss of production capability, financial loss or damage to equipment, property or environment. ....

Overall assessment

(h) Risks of harm to any person, property or environment is similar to that with the average pressure vessel provided chances of failure of Items (c) and (g) are kept rare or unlikely or ....

Risk level with controls in Item C4 is assessed at: ....

C4 RISK CONTROL (i.e. controls adopted to ensure overall risk in Paragraph C3.4 is acceptable).

(a) Vessel (and protective devices and controls): design, manufacture, tests, inspection comply with Australian Standards or equivalent. ....

(b) Vessel installation complies with AS 3892 or AS/NZS 1425, or equivalent. ....

(c) Operation, care and maintenance by competent persons and comply with AS 3873 or equivalent. ....

(d) Vessel and system are subjected to periodic checks in accordance with AS/NZS 3788, AS/NZS 1425, AS 2327 or equivalent. ....

(e) Special additional controls. ....

C5 COMMENT

....

....

....

C6 CERTIFICATION:

Supplied by Designer/manufacturer — Reviewed and completed by Owner/or Representative

Signature or Stamp .... *Date*.................... *Signature or Stamp* .... *Date*..........................

C7 MANAGEMENT REVIEW *(At least every 5 years). Signature* .... *Date:*....................................

C8 REVISIONS			*By*	*Approved*	*Date*
Revision	A	....	....	....	....
Revision	B	....	....	....	....
Revision	C	....	....	....	....

FIGURE C1 SAMPLE OF A RISK ASSESSMENT FORM

APPENDIX D
CORROSION PREVENTION
(Informative)

D1 GENERAL

Only general provisions have been made in this Standard for protection against corrosion, erosion and the like, (see Clause 3.2.4 and 2.5.3.3). Detailed information on this subject is beyond the scope of this Standard but opportunity is taken in this Appendix to provide some additional data and to suggest good practice in the selection of appropriate corrosion allowance.

D2 DATA ON CORROSION RESISTANCE

The corrosion resistance of metals listed in Table B1 and linings as permitted by Clause 3.2.4.4 vary considerably from material to material depending on the environmental conditions. Selection of materials with suitable corrosion resistance is vital in design.

Where actual prior experience is not available on corrosion resistance, information may be obtained from the material manufacturer or from sources such as the following:

1 American Petroleum Institute, RP 581 *Risk-based inspection technology*.

2 American Society for Metals, *Metals Handbook Volume 13, Corrosion*.

3 Deutsches Institut Fur Normung, DIN 50 929 *Probability of corrosion of metallic materials when subject to corrosion from the outside*, 1985.

4 National Association of Corrosion Engineers, International Publication 35103 *External Stress Corrosion cracking of Underground Pipelines*, 2003.

5 AS/NZS 2312*, Guide to the protection of iron and steel against exterior atmospheric corrosion*.

6 AS 2832.1, *Cathodic protection of metals, Part 1: Pipes and cables*.

D3 SUGGESTED GOOD PRACTICE REGARDING CORROSION ALLOWANCE

D3.1 General

Pressure vessels may be classified from a corrosion standpoint into one of the following groups:

(a) Vessels for which corrosion rates may be established definitely from information available to the designer covering the chemical characteristics of the substances they are to contain. Such information may, in the case of standard commercial products, be obtained from published sources, or where special processes are involved, from reliable records compiled from results of previous observations by the user or others under similar conditions of operation.

(b) Vessels in which corrosion rates, while known to be relatively high, are either variable or indeterminate in magnitude.

(c) Vessels in which corrosion rates, while indeterminate, are known to be relatively low.

(d) Vessels in which corrosion effects are known to be negligible or entirely absent.

D3.2 Predictable rate

Where the rate of corrosion is closely predictable, additional metal thickness over and above that required for the initial operating conditions should be provided at least equal to the expected corrosion loss during the design life of the vessel.

D3.3 Unpredictable rate

When corrosion effects are indeterminate prior to design of the vessel, although known to be inherent to some degree in the service for which the vessel is to be used, or when corrosion is incidental, localized, and/or variable in rate and extent, the best judgment of the designer must be exercised. A reasonable maximum excess shell thickness should be established at least equal to the expected corrosion loss during the desired life of the vessel, at the same time bearing in mind the provision of Paragraph D3.4, which will, in most cases, govern such vessels. For all vessels coming under this classification a minimum corrosion allowance of 1 mm should be provided unless a protective lining is employed. This lining, whether attached to the wall of a vessel or not, is not to be included in the computed thickness for the required shell thickness.

D3.4 Determination of probable corrosion rate

For new vessels and vessels for which service conditions are being changed, one of the following methods should be employed to determine the probable rate of corrosion from which the remaining wall thickness at the time of the inspection can be estimated:

(a) The corrosion rate established by accurate data collected by the owner or user on vessels in the same or similar service, should be used as the probable rate of corrosion.

(b) If accurate measurements are not available, the probable rate of corrosion may be estimated from experience of vessels in similar service.

(c) Where the probable corrosion rate cannot be determined by either of the above methods, thickness measurements should be made after 1000 h, or of other practical period of use, and subsequent sets of thickness measurements should be taken after additional similar intervals. If the probable corrosion rate is determined by this method, the rate found while the surface layer was present may not be applicable after the surface layer has disappeared.

D3.5 No corrosive effect

In all cases where corrosion effects can be shown to be negligible or entirely absent, no excess thickness need be provided.

D3.6 Corrosion inspection

Where a vessel goes into corrosive service without previous service experience, it is recommended that service inspections be made at frequent intervals until the nature and rate of corrosion in service can be definitely established (see Paragraph D3.4). The data thus secured should determine the subsequent intervals between service inspection and the probable safe operating life of the vessel. See also AS/NZS 3788 for information.

APPENDIX E

INFORMATION TO BE SUPPLIED BY THE PURCHASER TO THE DESIGNER/MANUFACTURER

(Informative)

E1 GENERAL

The technical input data shown in Paragraphs E2 to E7 and Figure E1 are necessary to enable the design, manufacture and supply of vessels that will comply with this Standard, satisfy the Purchaser's requirements, and perform safely and as-intended in service. This data is the subject of agreement between the Purchaser and Manufacturer.

Provision of technical input data should occur not later than the placement of the order. Where technical input data has not been supplied by the purchaser, the designer/manufacturer should advise the purchaser of all assumptions, changes and additions, to obtain agreement. This agreement should be obtained before the order or contract is accepted, to avoid delays and/or additional costs.

For ready-made or stock vessels, the designer/manufacturer assumes the role of the purchaser for the purpose of providing the technical input data specified in this Appendix.

The ‘designer’ of the vessel may include individual process designers, thermal designers, and designers for flow, piping and instrumentation, material, mechanical, foundations, safety devices and systems etc, and also other personnel.

E2 DESIGN

To enable the vessel to be designed in accordance with the minimum requirements (Clause 3.1.2), the following information should be supplied by the purchaser:

NOTE: Where the purchaser is responsible for the design (see Clause 3.1.2) some or all of these items may, by agreement, be omitted.

(a) Size and overall dimensions.

(b) Number, size, location and type of connections and openings.

(c) Type and mode of support.

(d) Design pressure and design temperature.

(e) Operating pressure and operating temperature, and if vessel is to operate below 20°C, the design minimum temperature and coincident pressure.

(f) Number of operating cycles expected from intended service of vessel.

(g) Material to be used and corrosion allowance (if equivalent materials are to be used this should be stated—see Clause 2.3).

NOTE: Information should include the name and/or chemical composition, the MSDS, and Dangerous Goods classification.

(h) Classification of vessel (see Clause 1.7).

(i) Nature of contents and type of gas, if vessel is to be used for liquefied gas storage.

(j) Statement whether vessel is to be used as a transportable vessel.

(k) Any significant loads to be applied to nozzles or other parts of vessel (see Clause 3.19.10.1(b)).

(l) Relevant environmental data to allow wind and seismic loading to be determined.

In addition to the minimum requirements of this Standard, the purchaser may require other features to be incorporated. These may rule out permissible alternatives as allowed by the Standard; require a higher quality workmanship; or require optional features to be incorporated. The following additional items may then have to be considered:

(i) Special tolerances on dimensions and machined surfaces.

(ii) Specific weld details.

(iii) Surface treatment and finishes internal and external.

(iv) Insulation—cold or hot—as required.

(v) Additional heat treatments.

(vi) Specific welding procedures to be used.

(vii) Specific inspection or testing techniques to be used, e.g. ultrasonic or magnetic particle examination.

(viii) Supply and installation of instruments, valves, safety valves and the like (see Clause 8.1.1).

(ix) Specific details on flanges, flange to nozzle connections, nozzle to shell connections, tubeplate to shell connections, and the like.

(x) Lifting lugs and associated reinforcements.

(xi) Limitation on weight (e.g. transport vessels).

(xii) Others.

E3 VERIFICATION OF DESIGN

Where the vessel is designed by the manufacturer, the purchaser should ensure that the manufacturer obtains the design verification as required. The purchaser should also state whether the design, specifications and drawings made by the manufacturer are to be accepted by the purchaser prior to start of manufacture.

NOTE: Where the vessel is designed by the purchaser it is expected that the purchaser will himself be responsible for ensuring appropriate design verification.

E4 INSPECTIONS

The purchaser should specify on the order any additional inspections required to be made and the stages at which these are to be carried out.

E5 TESTING

Where special tests, such as pneumatic testing (see Clause 5.11), corrosion testing (see Clause 5.17), leak testing (see Clause 5.13.4) and similar, are required, these shall be specified.

E6 DISPATCH

The purchaser should specify on the order, any particular requirements regarding cleaning, sealing, transportation and protection during transportation of vessel (see Section 9).

E7 CERTIFICATION AND DOCUMENTATION

The purchaser should specify any data required from the manufacturer (see Appendix F).

Customer/Purchaser		
No. Off	Order No.	Doc No.
Manufacturer		
Item No.	Job No.	Page 1 of

Vessel Description	Sketch No.	PandID No.
Function (e.g. HEX, receiver, storage, reactor, separator)		
Location of installation		
Construction Standard(s)		Vessel Class [1.7]
Hazard level (to AS 4343)		

Service conditions (normal and reasonably foreseeable, including upsets, startup, shut down)		
Overall shape and size	Ends	
Volume *(V)* L	Surface area (*A*) m^2	
D_i mm	*H* mm	*L* mm
Tube diam. mm	Tolerances	
Performance: Flow kg/h	Heat transfer kW	Liquid level
Pressure drop kPa	Velocity limit m/s	
Contents: Name	Type (lethal etc)	
Nature: Sour	Hydrogen p.press kPa	Erosive
Pressures (MPa): Max. working	Min. working	Margin for SV
Design pressure (*P*)	Test P	
Temperatures: Max.working °C	Min. working	
Max local difference °C	Design temperature (*T*)	
Design life: No of cycles (fatigue)	Hours (creep)	
Years (corrosion)	Indefinite	
Specific materials to suit service[2.3]: Shell(s)	Tubes	
Fittings		
Nozzles: No.	Size mm	Location	Material
Function	Flow direction	
Connections: Type	Flange rating	Material
Gaskets	Bolting	

NOTES: See end of table

FIGURE E1 (in part) TECHNICAL DATA FOR DESIGN, MANUFACTURE AND SUPPLY

Corrosion and erosion: Type	Corrosion allowance mm	Fouling	
Deposits			
Attachments: Supports - type	No.	Location	
Lifting: lugs	Trunnions		
Ladders	Platforms	Davits	
Internals			
Firing arrangements for fired vessels			
Special welds for: Corrosion	Erosion	Hydrogen	
Fatigue	Draining	Other	
Access for cleaning, inspection and maintenance:			
Openings: No.	Size	Location	
Draining:	Drains	Vents	
Gaps	Slope		
Insulation: Intern	Extern	Noise	Hot
Cold	Fire	Thickness mm	
Surface treatment: Intern	Extern	Preparation	Finish
Paint/coatings/liners			
Protective devices (safety valves etc): Type	No.	Material	
Impingement *p*	Earthing		
In-service management: Operation and maintenance to AS 3873			
Inspection to AS/NZS 3788	Security		

Location (factors influencing the design, manufacture and supply)			
Local conditions: Wind	Rain	Hail	Snow
Flood	Wave	Seismic	Earth settlement
Dust	Temperature	Altitude	Remote
Near- sea	Industry	Dangerous material	Public
Installed: In open	Indoors	Buried	
Mounded	Shaded	To AS 3892	
Transport: To site only			
For transport of fluid by: Road	Rail	Sea	Air
Special: Foundations	Barriers	Clearances	
Facilities			
Applicable regulations: OHS	Major hazard	Public safety	
Transport	Environment		

FIGURE E1 (in part) TECHNICAL DATA FOR DESIGN, MANUFACTURE AND SUPPLY

Loads (and actions) i.e. forces and moments in-service and during fabrication, test, transport and installation				
Pressure MPa: Max.	Min.	Internal	External	
Design	Head	Vacuum	Calculation *P*	
Mass of vessel (kg) : Empty	With insulation and attachments			
With contents	With water			
Live loads: Wind	Snow	Flood	Wave	
Seismic	Earth settlement	Transport (G)		
Special nozzle loads [3.19.10.1a] from: Piping				
Flow or discharge reactions				
Support loads: Vessel				
From attached service or maintenance equipment				
Special loads: Expansion and contraction in vessel	Collision			
Projectiles	Dropping	Blast		
Fluctuation of loads: Cycles	Sloshing			
Shock (water hammer)	Vibration			
Lifting loads in: Manufacture	Installation			
Use	Maintenance			
Hydro-test loads				
Test positions: shop	site	vertical	horizontal	

Manufacture and test			
Type of construction: Welded	Specific w. procedure		
Brazed	Forged	Cast	Other
Heat treatment: Type	When		
Special NDE [2.8] or other tests			
Pressure test: Hydro	Pneum.	Leak	Medium
Special draining/drying after hydro			

FIGURE E1 (in part) TECHNICAL DATA FOR DESIGN, MANUFACTURE AND SUPPLY

Purchase/Supply requirements		
Conformity Assessment: Cert. QM System		
Independent Inspection body		
Registration required: Design : Yes	No	with
Vessel : Yes	No	with
Valves, fittings, insulation: Supply	Installation	
Vessel markings and nameplates		
Special requirements: Dispatch/delivery [Section 9]		
Documentation [App. F]		

Other Remarks			
Prepared by	Date	Checked by	Date
Customer Noted	Date	ISSUE No.	

NOTES:

1 Equivalents to this form may be used. Numbers in [ ] refer to Clauses in AS 1210. ✓ = Yes; x = No.

2 This Form also provides a check-list of most initial data needed to suitably construct most vessels and for MDR.

3 Additional data on fluids and material properties may be needed from the Purchaser or Manufacturer.

4 For multi-chamber vessels e.g. heat exchangers and jacketed vessels, data for each chamber is necessary, and another page may be required.

5 Critical, unusual and special data should be highlighted to avoid oversight.

FIGURE E1 (in part) TECHNICAL DATA FOR DESIGN, MANUFACTURE AND SUPPLY

APPENDIX F

INFORMATION TO BE SUPPLIED BY THE DESIGNER/MANUFACTURER

(Informative)

The following information should be supplied:

(a) *By the designer to the manufacturer*:

(i) General arrangement and other drawings necessary for vessel manufacture.

(ii) Information on the material and method (e.g. heat treatment, examinations, tests and inspection) necessary for manufacture.

(iii) Design information to permit completion of the manufacturer's data report.

(iv) Information on those special or unusual requirements not covered by AS 3892, AS 3873 and AS/NZS 3788, e.g. special transport, installation, commissioning, operation, maintenance, inspection and testing (e.g. special tests when approaching creep or fatigue design life).

(v) If specified, a risk assessment (see Paragraph C2).

(b) *By the designer to the design verification body (where required):*

(i) Information in Items (a)(i)–(iii) above.

(ii) Design calculations.

(iii) Other data necessary for the purposes of design verification.

(c) *By the manufacturer to the fabrication inspection body (where required):*

NOTE: This data may be alternatively made available to the inspector.

(i) Information in Item (a)(i)–(ii) above.

(ii) Material test certificates, qualified welding procedures, welder qualification, production test results, heat treatment certificates, report on non-destructive examination and other applicable requirements of AS 4458.

(d) *By the manufacturer to the purchaser*:

(i) The manufacturer's data report (and other information agreed by the parties concerned at time of placement of order), see AS 4458.

(ii) Such additional data as is required by the purchaser on the order, e.g. calculations, drawings, specifications, risk statement and operating instructions.

(iii) The information in Item (a)(iv).

Unless otherwise agreed by the parties concerned—

(A) the above information should be supplied in the English language;

(B) any required design verification should be done before commencement of manufacture; and

(C) any required design registration should be arranged by the designer.

APPENDIX G

FAILURE MODES

(Normative)

Vessels shall be designed and manufactured to avoid the failure modes given in Table G1, during manufacture, transport, installation, operation and maintenance under specified and reasonably feasible service conditions.

Compliance with AS 1210 is intended to ensure the above but the designer and manufacturer may need to supplement requirements in various clauses to ensure compliance, e.g. limits on deflection for special service conditions or to ensure engagement of quick actuation closures.

'Failure mode' is the basic manner or mechanism of the failure or deterioration process. It is also known as 'damage mechanism'.

TABLE G1

FAILURE MODES OF METALS

Failure modes		Limit state	Action type	
			Duration	Application
1	**CORROSION**			
1.1	General and uniform corrosion (See Note)	S	Long term	Single
1.2	Localized corrosion	S	Long term	Single
	▪ Small			
	▪ Pitting			
	▪ Crevice			
1.3	Galvanic corrosion	S	Long term	Single
1.4	Velocity-related corrosion	S	Long term	Single
	▪ Erosion/abrasion			
	▪ Cavitation			
	▪ Fretting			
1.5	Intergranular corrosion	S	Long term	Single
	▪ Weld decay			
	▪ Exfoliation			
1.6	Dealloying corrosion	S	Long term	Single

(*continued*)

TABLE G1 (*continued*)

Failure modes		Limit state	Action type	
			Duration	**Application**
1.7	Corrosion induced cracking (environmentally assisted) ▪ Corrosion fatigue ▪ Stress corrosion cracking (SCC), by chloride or caustic ▪ Hydrogen induced cracking (HIC) ▪ Stress oriented hydrogen induced cracking (SOHIC) ▪ Hydrogen blistering ▪ Liquid metal embrittlement (LME) ▪ Microbiologically induced corrosion (MIC)	U	Long term	Single (Cyclic)
1.8	High-temperature corrosion ▪ Oxidation/scaling ▪ Carburization ▪ Decarburization ▪ Internal oxidation ▪ Metal dusting ▪ Hydrogen sulphide cracking ▪ Hydrogen attack	S	Long term	Single
2	**HIGH-TEMPERATURE METALLURGICAL DAMAGE (See Note)**			
2.1	Spheroidization (softening)	S	Long term	Multiple
2.2	Temper embrittlement	S	Short/Long	Single/Multiple
2.3	885F embrittlement	S	Short term	Single
2.4	Graphitization	S	Long term	Multiple
2.5	Sigma phase embrittlement	S	Short/Long	Single/Multiple
2.6	Reheat cracking	U	Short/Long	Single/Multiple
2.7	Solidification cracking	U	Short term	Single
3	**CREEP (See Note)**			
3.1	Creep distortion	S	Long term	Single/Multiple
3.2	Creep cracking	U	Long term	Single/Multiple
3.3	Creep rupture (through wall)	U	Long term	Single/Multiple
3.4	Creep fatigue	U	Long term	Cyclic
3.5	Creep buckling	U	Long term	Single/Multiple
4	**BRITTLE FRACTURE (See Note)**			
4.1	Ferritic steel at notches at low temperature	U	Short term	Single
4.2	Low ductility material: Cast iron, glass, graphite, some plastics	U	Short term	Single

(*continued*)

TABLE G1 (*continued*)

Failure modes		Limit state	Action type	
			Duration	**Application**
5	**FATIGUE (See Note)**			
5.1	Mechanical fatigue (high/low cycle)	U	Long term	Cyclic
5.2	Vibration induced fatigue	U	Long term	Cyclic
5.3	Thermal fatigue	U	Long term	Cyclic
5.4	Thermal shock	U	Long term	Cyclic
(see also Corrosion fatigue, Creep fatigue)				
6	**EXCESSIVE DEFORMATION (elastic or plastic) (See Note)**			
6.1	Thermal deformation by overheating	S	Short term	Single
6.2	Over-straining (e.g. by over pressurization, overload, earthquakes, earth settlement and wind)	S	Short term	Single
6.3	Deformation causing leakage at mechanical joints	S	Short/Long	Single
6.4	Progressive plastic deformation, ratchetting, incremental collapse	U	Short term	Multiple
6.5	Mechanical gouging or dents	S	Short term	Single
7	**DUCTILE FRACTURE (See Note)**			
7.1	Ductile rupture following gross plastic deformation (includes ductile tearing, plastic collapse, plastic instability and bursting)	U	Short term	Single
7.2	Lamellar tearing	S or U	Short term	Single
8	**INSTABILITY (BUCKLING OR COLLAPSE) (See Note)**			
8.1	Elastic buckling	S	Short term	Single
8.2	Plastic or elastic plastic buckling	S	Short term	Single
8.3	Overturning	S	Short term	Single
	(See also Creep buckling)			
9	**LEAKAGE (See Note)**			
9.1	Through defective joints and materials	S	Short/Long	Single/Multiple
(see other modes leading to leakage, e.g., corrosion, fracture, cracking and excessive deformation)				
10	**SUSTAINED LOAD CRACKING (See Note)**			
In 6000 series aluminium alloys		U	Long term	Multiple
11	**COMBINATION OF ABOVE (See Note)**			
Mixed modes, e.g., corrosion or fatigue leading to brittle fracture		S or U	Short/ Long	Single/Multiple

LEGEND:

S = Serviceability limit state, i.e., structural condition resulting in excessive deformation, deflection or leakage but not compromising safety

U = Strength or Ultimate limit state, i.e. structural condition resulting in burst, collapse or danger to people

NOTE: May be *U* when major risk results from leakage, etc.

APPENDIX H

STRESS INTENSITY CLASSIFICATION AND LIMITS

(Normative)

H1 GENERAL

This Appendix provides guidance on the classification and limits of stress intensities derived using linear elastic analysis. This is required to interpret results from rigorous stress analysis, e.g. linear elastic finite element analysis.

These classifications and limits shall not be used with the results of non-linear finite element analysis.

H2 STRESS INTENSITY CLASSIFICATION

Stress intensities shall be classified by their nature (primary, secondary, peak) and distribution (membrane, bending, general, and local) as follows:

(a) *Stress intensity*—the equivalent uniaxial stress with respect to yielding for multiaxial stresses at a point. The Tresca stress intensity is used in this standard and numerically equates to the largest algebraic difference between the three principal stresses at a point, e.g. if the three principal stresses at a point are −100, +10 and +140 MPa, the Tresca stress intensity would be 240 MPa and would initiate yielding in a material having a yield strength of 240 MPa. The Tresca stress intensity at a point equates to twice the maximum shear stress at the point (see also Paragraph H4).

(b) *Primary stress intensity*—is the component of stress intensity whose magnitude is unaltered by yielding, that is non-self-limiting, and load controlled. It is a stress intensity produced by mechanical loadings only and is so distributed in the structure that no redistribution of load occurs as a result of yielding. It is a stress developed by the imposed loading which is necessary to satisfy the simple laws of equilibrium of external and internal forces and moments. Primary stresses that considerably exceed the yield stress will result in failure, or at least in gross distortion. A thermal stress is not classified as a primary stress. Primary stress is divided into 'general' and 'local' categories.

    Examples of general primary stresses are —

    (i) the average through-thickness stress in a circular, cylindrical or spherical shell due to internal pressure or to distributed live loads; and

    (ii) bending stress in the central portion of a flat head due to pressure.

(c) *Secondary stress intensity*—is the component of stress intensity whose value and distribution is altered by yielding, that is self-limiting, and deflection controlled. It is developed by the constraint of adjacent parts or by self-constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur, and failure from one application of the stress is not to be expected.

    Examples of secondary stress are bending stress at a gross structural discontinuity, and differential thermal expansion.

(d) *Peak stress intensity*—is a highly localized increment in stress intensity associated with a stress concentration such as a weld toe. The combination of primary + secondary + peak stress intensities at a point is required for fatigue analysis, but can otherwise be ignored, excepting for fracture mechanics and brittle fracture considerations. The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or brittle fracture. A stress that is not highly localized falls into this category if it is of a type which cannot cause noticeable distortion.

Examples of peak stress are—

(i) the differential thermal expansion stress in the austenitic steel cladding of a carbon steel vessel;

(ii) the surface stresses in the wall of a vessel or pipe produced by thermal shock; and

(iii) the stress at a stress concentration, e.g. a weld toe.

(e) *Local primary stress intensity*—cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a primary or a discontinuity effect or both, produces excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress. A stressed region may be considered as local if the distance over which the stress intensity exceeds $1.1f$ does not extend in the meridional direction more than $0.5\sqrt{(Rt)}$ and if it is not closer in the meridional direction than $2.5\sqrt{(Rt)}$ to another region where the limits of general primary membrane stress are exceeded. (R is the distance measured along the perpendicular to the surface from the axis of revolution of the vessel to the midsurface of the vessel; t is the wall thickness at the location where the general primary membrane stress limit is exceeded.)

An example of a primary local stress is the membrane stress in a shell produced by an external load and moment at a permanent support or at a nozzle connection.

(f) *Membrane stress intensity*—is the mean stress intensity through the thickness under consideration.

(g) *General stress intensity*—the stress intensity remote from discontinuities, e.g. nozzles and stress concentrations, e.g. weld toes.

(h) *Local stress intensity*—the stress intensity remote from stress concentrations but not remote from discontinuities.

(i) *Bending stress intensity*—the component of stress intensity at a point in a plate or shell after subtraction of the membrane stress intensity.

(j) *Structural discontinuity*—a source of stress intensification which affects a relatively large portion of a vessel and which has a significant effect on the overall stress pattern on the vessel.

Examples of structural discontinuities are head-to-shell and flange-to-shell junctions, nozzles and junctions between shells of different diameters or thicknesses.

(k) *Stress concentration*—source of peak stress which affects a relatively small volume of material but which does not have a significant effect on the overall stress pattern or on the structure as a whole.

Examples of stress concentrations are weld toes, small fillet radii, roots of partial penetration welds, and root radii of screw threads.

(l) *Normal stress*—the component of stress on a plane that is normal to the plane, such as compressive or tensile stress (sometimes referred to as 'direct stress'.)

(m) *Shear stress*—that component of stress on a plane that is tangential to the plane.

(n) *Principal stress*—a normal stress on a plane (a principal plane) that plane having a zero shear stress applied to it. At a point there are always three principal stresses on orthogonal planes, which can be determined by the Transformation of Stress Equations or their Mohr's Circle graphical embodiment, e.g. the inner and outer surfaces of a shell are typically principal planes, having usually only normal stress in the form of fluid pressure applied to them.

(o) *Bending stress* (σ)—the component of stress, proportional to the distance from the section mid-plane of a solid cross section. This is maximum on the surface and given by—

$$\sigma_b = \frac{6}{t^2} \int \sigma X \, dX \qquad \text{. . . H2}$$

$$= \frac{6}{t^2} M_1 \text{ maximum bending stress for linear distributions}$$

where

σ = bending stress, in megapascals

σ_b = maximum bending stress, in megapascals

t = thickness of the solid section, in millimetres

X = distance from the section mid-plane, in millimetres

M_1 = bending moment per unit edge length, in newton millimetres per millimetre

H3 EXAMPLES OF STRESSES AND THEIR CLASSIFICATIONS

Table H1 and Figure H1 have been included to guide the designer in establishing stress categories for some typical cases and stress intensity limits for combinations of stress categories. There will be instances when reference to definitions of stresses will be necessary to classify a specific stress condition to a stress category.

TABLE H1

CLASSIFICATION OF STRESSES FOR SOME TYPICAL CASES

Vessel component	Location	Origin of stress	Type of stress	Classification
Cylindrical or spherical shell	Shell plate remote from discontinuities	Internal pressure	General membrane (average stress intensity through plate thickness)	f_m
		Axial thermal gradient	Membrane	f_g
			Bending	f_g
	Junction with head or flange	Internal pressure	Membrane	f_L
			Bending	f_g
Any shell or end	Any section across entire vessel	External load or moment, or internal pressure	General membrane (averaged across full section). Stress component perpendicular to cross-section	f_m
		External load or moment	Bending across full section. Stress component perpendicular to cross-section	f_m
	Near nozzle or other opening	External load or moment, or internal pressure	Local membrane	f_L
			Bending	f_g
			Peak (fillet or corner)	f_p
	Any location	Temperature difference between shell and head	Membrane	f_g
			Bending	f_g
Dished end or conical end	Crown	Internal pressure	Membrane	f_m
			Bending	f_b
	Knuckle or junction to shell	Internal pressure	Membrane	f_L*
			Bending	f_g
Flat end	Centre region	Internal pressure	Membrane	f_m
			Bending	f_b
	Junction to shell	Internal pressure	Membrane	f_L
			Bending (see Note)	f_b
Perforated end or shell	Typical ligament in a uniform pattern	Pressure	Membrane averaged through cross-section	f_m
			Bending (averaged through width of ligament, but gradient through plate)	f_b
			Peak	f_p
A2	Isolated or atypical ligament	Pressure	Membrane	f_g
			Bending	f_p
			Peak	f_p

(*continued*)

TABLE H1 (*continued*)

Vessel component	Location	Origin of stress	Type of stress	Classification
Nozzle	Cross-section perpendicular to nozzle axis	Internal pressure or external load or moment	General membrane (averaged across full section). Stress component perpendicular to section	f_m
		External load or moment	Bending across nozzle section	f_m
	Nozzle wall	Internal pressure	General membrane	f_m
			Local membrane	f_L
			Bending	f_g
			Peak	f_p
		Differential expansion	Membrane	f_g
			Bending	f_g
			Peak	f_p
Cladding	Any	Differential expansion	Membrane	f_p
			Bending	f_p
Any	Any	Thermal gradient through plate thickness	Bending	f_p†
	Any	Any	Stress concentration (notch effect)	f_p

* Consideration should also be given to the possibility of buckling and excessive deformation in vessels with large diameter/thickness ratio.

† Consider possibility of thermal stress ratchet

NOTE: The edge bending moment may be classified as a secondary stress (f_g) only where the edge bending moment is not required to maintain the bending stress in the centre of the flat end within acceptable limits after shakedown

Stress category	Primary			Secondary	Peak
	General	Local	Bending		
Description (for examples, see Table H1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations. (See Note 2)	(1) Increment added to primary or secondary stress by a concentration (notch). (2) Certain thermal stresses which may cause fatigue but not distortion of vessel shape. (See Note 2)
Symbol (Note 4)	f_m	f_L	f_b	f_g	f_p
Combination of stress components and allowable limits of stress intensities					

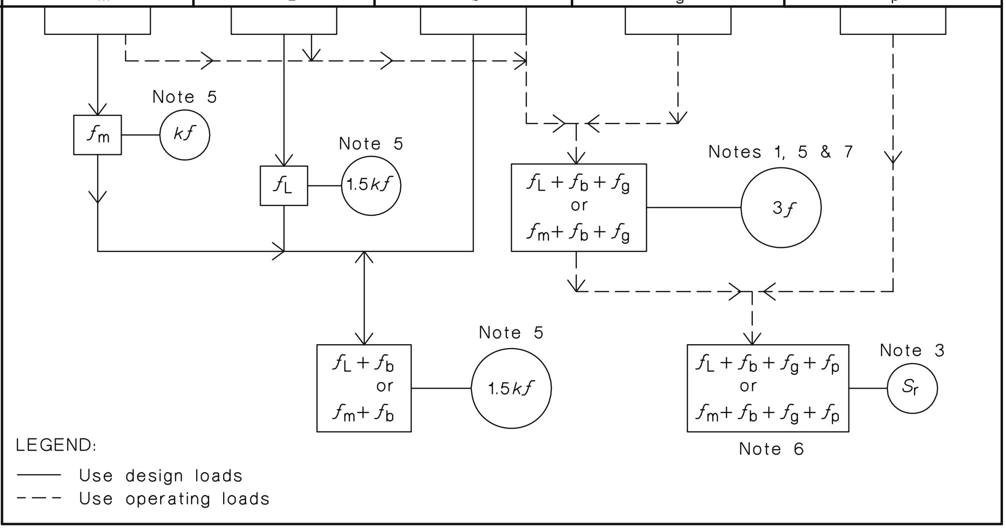


FIGURE H1 STRESS CATEGORIES AND LIMITS OF STRESS INTENSITY

NOTES TO FIGURE H1:

1 This limitation applies to the range of stress intensity. Where the secondary stress is due to a temperature excursion at the point at which the stresses are being analysed, the value of f is to be taken as the average of the f-values for the highest and the lowest temperature of the metal during the transient. Where part or all of the secondary stress is due to mechanical load, the value of f is to be taken as the f-value for the highest temperature of the metal during the transient.

2 The stresses in category f_g are those parts of the total stress which are produced by thermal gradients, structural discontinuities, etc, and do not include primary stresses which may also exist at the same point. It should be noted however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of f_m (or f_L) + f_b + f_g and not f_g alone. Similarly, if the stress in category f_p is produced by a stress concentration, the quantity p is the additional stress produced by the notch, over and above nominal stress. For example, if a plate has a nominal stress intensity f_1 and has a notch with a stress concentration factor K, then $f_m = f_1$, $f_b = 0$, $f_p = f_m(K - 1)$ and the peak stress intensity equals $f_m + f_m\,(K - 1) = Kf_m$.

3 S_r is obtained from the fatigue curves (Appendix M, Figures M3 and M6).

4 The symbols f_m, f_L, f_b, f_g and f_p do not represent single stresses at a point, but rather Tresca stress intensities.

5 The factor k is obtained from Table 3.1.6.

6 For fatigue analysis of weld toe stress concentrations it is permissible to use the geometric stress instead of ($f_L + f_b + f_g + f_p$) see Paragraph M7.3(b).

7 For primary plus secondary stress intensities the limit is set at $3f$. In a typical case f would equal two thirds of the yield strength, giving a limit for the combined primary plus secondary stress intensities of twice the yield strength. Therefore if primary plus secondary stresses are set to this limit then on the first loading cycle, yielding will take place. In an extreme example, where all the stress is secondary in nature (such as restrained thermal expansion in the absence of load or pressure related primary stresses) unloading after the first loading will result in favourable residual stresses equal to yield of the opposite sense to those generated during loading. On the second cycle ideally then the stress excursion will be from yield in one sense to yield in the other (i.e. elastically through a range of twice yield). In practice this process of 'shakedown' occurs over the first few cycles. It must be clearly understood that the use of the limit of $3f$

(typically twice yield strength) for the combination of primary plus secondary stress intensity is restricted to stresses derived from linear elastic analyses. Stresses derived from non-linear analyses (i.e. based on the real stress/strain curve for the material) may not be used with this limit, it being noted that (for example in an elastic perfect plastic material) stresses equal to twice yield strength can never be reached regardless of the strain applied. Similarly the limits for peak stresses (S_r) referred to here and in Appendix M are set out for exclusive use with the results of linear elastic stress analyses, and may not be used with the results of non-linear stress analyses.

H4 TRESCA OR MAXIMUM SHEAR STRESS CRITERION

H4.1 General

The maximum shear stress at a point is defined as one-half of the algebraic difference between the largest and the smallest of the three principal stresses. Thus if the principal stresses are σ_1, σ_2, and σ_3 and if $\sigma_1 > \sigma_2 > \sigma_3$ (algebraically) the maximum shear stress is $0.5\ (\sigma_1 - \sigma_3)$. The maximum shear stress criterion states that yielding in a multiaxially stressed component initiates when the maximum shear stress in the component reaches a magnitude equal to the maximum shear stress in the uniaxial tensile test specimen at the yield. In such a tensile test—

$$\sigma_1 = f_y;\ \sigma_2 = 0;\ \text{and}\ \sigma_3 = 0.$$

where

f_y = yield strength

and therefore the maximum shear stress is $0.5f_y$, and yielding occurs when—

$$0.5(\sigma_1 - \sigma_3) = 0.5f_y$$

By analogy this equation can be written as—

$$(\sigma_1 - \sigma_3) = f_y$$

and the left-hand side is then called the equivalent uniaxial stress, or reduced stress or combined stress or Tresca stress intensity or simply ‘stress intensity’. Thus the ‘stress intensity’ is defined as twice the maximum shear stress and is equal to the largest algebraic difference between any two of the three principal stresses and is directly comparable to yield strength values found from tensile tests.

H4.2 Application of the Tresca Criterion

Considering a thin cylinder shell subject only to internal pressure loading, the following membrane stress apply:

$$\sigma_1 = \text{hoop stress} = \frac{PD}{2t}$$

$$\sigma_2 = \text{axial stress} = \frac{PD}{4t}$$

$$\sigma_3 = \text{radial stress}$$
$$= -P \text{ at inner surface}$$
$$= 0 \text{ at outer surface}$$

where

D = inside diameter of cylinder, in millimetres

P = internal pressure, in megapascals

t = wall thickness of cylinder, in millimetres

The mean radial stress for thin shells can therefore be taken as—

$$(\sigma_{rad})\text{ mean} = \sigma_{mean} = \frac{-P+0}{2} = -0.5P$$

From the above equations it then follows that the primary general membrane stress intensity is governed only by σ_1 and σ_3 and becomes—

$$\begin{aligned} f_m &= \sigma_1 - \sigma_3 \\ &= \frac{PD}{2t} - (-0.5P) \\ &= \frac{PD}{2t} + 0.5P \quad \text{which must not exceed the design strength } f \end{aligned}$$

A1 thus $t = \dfrac{PD}{2f - P}$ which is Equation 3.7.3(1) with joint efficiency (η) equal to 1, and with $k = 1$ (Table 3.1.6).

where

f_m = primary general membrane stress intensity

D = inside diameter of cylinder, in millimetres

P = internal pressure, in megapascals

f = design strength, in megapascals

t = wall thickness of cylinder, in millimetres

If in addition to internal pressure the vessel part is subject to other loads, the absolute and the relative values of σ_1, σ_2 and σ_3 will vary and could lead to a wall thickness larger than that given by, say, Equation 3.7.3(1). Thus, if a slender cylinder is subject to bending, σ_2 could become larger than σ_1 and the calculated stress intensity would become equal to $\sigma_2 - \sigma_3$.

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H5 VON MISES CRITERION

While this Standard generally cites the Tresca criterion, the Tresca criterion is a linear approximation of the von Mises criterion. Accordingly, it is permissible for the purposes of stress analyses in this Standard to substitute von Mises stresses for Tresca stresses.

A2

APPENDIX I

FINITE ELEMENT ANALYSIS

(Normative)

I1 GENERAL

This Appendix gives two alternative methods of pressure vessel design using Finite Element Analysis (FEA). These two FEA based methods of establishing the integrity of the design of a vessel are alternatives to the rule based design methods given in Section 3 of this Standard. Compliance with the requirements of at least one of these three methods is sufficient to demonstrate the adequacy of the design of a vessel or its components. These two alternative FEA based methods are as follows:

(a) *Linear elastic FEA*

Linear analyses do not include the stress/strain properties of the material above the yield strength and as such always give results where stress and strain are related by Young's modulus and Poisson's ratio for stresses both above and below the yield strength. The output of such linear analyses shall be interpreted using the stress categories as described in Appendix H, and shall comply with the stress intensity limits given in Appendix H.

(b) *Non-linear FEA*

Non-linear analyses include the stress/strain properties of the material. The output of such non-linear analyses shall comply with the strain requirements of Paragraph I3.

In addition to strength requirements, stability performance may also be analysed using non-linear FEA, see Paragraph I4.

Also in addition to the strength requirements, vibration analysis may be carried out using either linear or non-linear FEA, see Paragraph I5.

Finite element stress analysis of pressure equipment should only be carried out by competent stress analysts who are also competent in the use of FEA.

I2 STRENGTH DESIGN BASED ON STRESSES FROM LINEAR ANALYSES

I2.1 Designs reliant on linear FEA

Designs reliant on linear FEA shall be evaluated using the stress categories and permissible stress intensities given in Appendix H, and in the case of fatigue life in accordance with Appendix M.

NOTE: The stress intensity limits given in Appendices H and M are *only* for use with the results of linear elastic stress analyses. The stress intensity limits given in Appendices H and M are likely to give highly unconservative results if used with stresses above yield that are generated from non-linear stress analysis (which is based on the actual stress/strain curve of the material rather than being based on the simple Young's modulus elastic relationship).

I2.2 Yield criteria

In general, the yield criteria referred to in this Standard is the Tresca Criterion, where the Tresca stress intensity at a point is the algebraic difference between the maximum and minimum principal stress at that point. However, Tresca stress is a linear approximation of the von Mises stress and, accordingly, it is permissible to use the von Mises stress intensity rather than the Tresca stress intensity when comparing stress intensities with the limits given in Appendix H. When the term 'stress' is used in this Standard it is to be understood to mean either Tresca or von Mises stress intensity unless otherwise specified.

A2

I2.3 Meshing technique

In all cases the meshing technique should ensure the following:

(a) Large elements are not adjacent to small elements. Element size should vary through the structure smoothly. The ratio of adjacent element sizes in regions of interest should not exceed 2:1.

(b) That the aspect ratio of elements is in the range 0.33 to 3.

(c) That four sided elements are used in preference to three sided elements, and higher order elements are used in preference to lower order elements.

(d) That structural discontinuities have sufficient elements to capture the local behaviour (e.g. a cylindrical shell has a characteristic length $L + 0.55\sqrt{Dt}$ and a hole in a plate has a characteristic length equal to its radius).

In such cases at least two quadratic elements or six linear elements within this characteristic length are required to capture local behaviour, where this is important.

(e) That benchmark standard results or established analytical methods are used to help verify the output. For example, membrane stresses and bending stresses can often be calculated at locations remote from discontinuities.

(f) That a mesh/grid having an element spacing that varies smoothly throughout the structure is selected.

(g) That boundary conditions (e.g. planes of symmetry and imposed loads) can be readily verified.

I2.4 Consistency and credibility of results

To ensure consistency and credibility FEA results shall be inspected using the following criteria:

(a) Output contours shall be free of local meshing anomalies such as scalloping, particularly in those areas of the model relied on for numerical values of stress used in assessing the integrity of the vessel or component.

(b) The deflection of the structure shall appear reasonable in shape and magnitude.

(c) The maximum variation in stress across any element, excluding those adjacent to singularities, shall not exceed the following:

Element order	Maximum stress variation
0	10%
1	20%
2	30%
>2	40%

(d) Singularities such as sharp internal corners at weld toes are acceptable, providing the stresses in the immediately adjacent elements are not relied on in the assessment of the vessel's integrity. Such sharp corners are of significance in fatigue analysis, however detailed modelling of same can be avoided by using the geometric stress method (see Appendix M).

I2.5 Stress distribution

The distribution of the components of stress across the thickness can be determined using the following equations:

Membrane stress $\sigma_{\mathrm{m}} = \frac{1}{t} \int \sigma \mathrm{d}x$ . . . I2(1)

A2

Bending stress $\sigma_b = \frac{1}{t^2} \int \sigma x \, dx$ . . . I2(2)

where x = distance from the mid plane thickness

For plate elements whose formulation assumes a linear stress distribution through the thickness these stress components are readily found from:

σ_m = mid-plane stress

σ_b = surface stress – mid-plane stress

I2.6 Stress evaluation

In order to evaluate the resulting stresses from linear FEA for non-buckling structures the stresses shall be classified in accordance with—

(a) their distribution through thickness (membrane and bending); and

(b) their nature, whether they are self-limiting (secondary) or non-self-limiting (primary).

The nature of the stress (whether self-limiting or not) shall be inferred using linear superposition by—

(i) separating mechanically induced stresses (e.g. from pressure) from known secondary stresses (e.g. thermal);

(ii) calculating and subtracting that component of stress in the vicinity of a structural discontinuity due to known stresses, which can be readily determined by simple analytical techniques, e.g. membrane pressure stresses and flat plate bending stresses; and

(iii) calculating that component of a stress due to mismatch, e.g. cladding, interface or other self-limiting effects.

When stresses from a linear elastic FEA have been appropriately classified as above, they can be compared to the stress category limits in Appendix H using the basic design strength, f.

I3 STRENGTH DESIGN BASED ON STRAINS FROM NON-LINEAR ANALYSES

I3.1 General

This Paragraph I3 provides a means to prove the design integrity of a vessel or pressure component with respect to strength, using non-linear finite element analysis.

NOTE: While this Paragraph specifically addresses design for strength using non-linear finite element analysis, many vessels will also have deformation related serviceability limits that may be analysed concurrently with the strength requirements.

I3.2 Requirements

For strength design based on strains from non-linear analyses, the following applies:

(a) The finite element analysis shall be non-linear, including both non-linear geometry and non-linear material properties (see Paragraph I3.3) excepting fatigue analysis [see Item (h)].

(b) All reasonably foreseeable significant loads and load combinations shall be analysed including normal design conditions, start up, shut down, upset conditions, thermal loading, wind and seismic loading, with the vessel in its corroded condition.

(c) Non-linear analysis shall be used to determine the vessel shape after the hydrostatic test. The hydrostatic test pressure shall be no less than that determined by

A2 Equations 5.10.2(a), 5.10.2(b) and 5.10.2(d). The resulting calculated strains shall be limited to the following values:

(i) Limit the inelastic strain (see Note 1) remote from discontinuities and peak strain regions at the hydro test pressure and hydro test temperature to be less than 1% for vessels other than cold stretched austenitic stainless steel vessels and to be less than 5.2% for cold stretched austenitic stainless steel vessels (see Paragraph L5.3, Appendix L).

(ii) Limit the inelastic strain at all locations excluding peak strain locations at the hydro test pressure and at the hydro test temperature to be the lesser of 5% and one third of the material's failure elongation (see Note 2) for vessels other than cold stretched austenitic stainless steel vessels and to be the lesser of 25% and one third of the material's failure elongation for cold stretched austenitic stainless steel vessels.

If the hydrotest simulation results in greater inelastic strains, the design shall be revised until it is in compliance with the strain limits given above.

(d) All reasonably foreseeable significant loads and load combinations applied in service after hydrostatic testing shall result in elastic only strain excluding peak strain regions (see Notes 3 and 4).

(e) 1.5 times all reasonably foreseeable significant loads and load combinations applied in service after hydrostatic testing shall result in elastic only strain remote from discontinuities and peak strain regions.

(f) For those materials having a stress/strain curve in which the magnitude of $R_m/2.35$ (or for austenitic steels $R_m/2.5$) is less than $R_e/1.5$ all reasonably foreseeable significant loads and load combinations applied in service after hydrostatic testing multiplied by 2 (2.15 for austenitic steels) shall be capable of being applied to the vessel without causing collapse or bursting (see Note 5).

(g) Where the hydrostatic test and service load analyses of the vessel are carried out at the same temperature and that temperature differs from the actual service temperature, for the purposes of the service load analyses the service loads shall be multiplied by the ratio of the design strength at hydrostatic temperature to the design strength at the service temperature.

(h) For those vessels subject to cyclic loading, fatigue analysis shall be carried out using linear elastic stress/strain material properties according to Appendix M based on the vessel shape after hydrostatic testing as determined by non-linear analysis.

NOTES:

1 It is necessary that the non-linear FEA software used be capable of giving a contour plot of inelastic strain (often referred to as plastic strain) in order that the inelastic strains resulting from the hydrostatic test can be verified as being within the permissible limits.

2 For the purposes of Paragraph I3.2 Item (c)(ii) the failure elongation is the engineering strain at failure taken from the engineering stress/strain curve used as the basis for the true stress/strain curve employed in the non-linear analysis.

3 The non-linear FEA of the hydro test should result in the unloaded empty vessel having: the modified shape, the residual stress distribution, and the strain hardening distribution resulting from the hydro test. Starting from this post hydro test condition enables the subsequent non-linear FEA of the service loads to fully capture the benefits of stretching during hydro testing. It is necessary that the nonlinear FEA software used be capable of giving a contour plot of inelastic strain in order that elastic action resulting from the service loads (at service temperature) can be verified.

4 For those cases where the service loads result in secondary stresses not shaken down to elastic action during the hydrostatic test it is permissible to apply the service loads more than once after the hydrostatic test during the nonlinear analysis to demonstrate elastic only action

A2 (excluding peak strain locations) resulting from the service loads. That is for example the non-linear FEA would comprise the following loading sequence:

Step 1 hydrostatic pressure applied and removed.

Step 2 service loads applied and removed.

Step 3 service loads applied a second time.

Step 4 service loads are increased by a factor of 1.5.

With elastic only strain (including at discontinuities but excluding peak strains) demonstrated to result from Step 3 (Step 2 ignored) and elastic only strain demonstrated to result from the combination of Steps 3 and 4 remote from discontinuities for compliance.

For fatigue analysis, the model shape may be saved after Step 1 and a separate analysis carried out using the fatigue loading with linear elastic material properties.

5 R_e and R_m are to be taken from the engineering stress/strain curve used as the basis for the true stress/strain curve employed in the non-linear analysis. The absence of collapse or bursting can be demonstrated by convergence to a solution in which strains have not exceeded the maximum strain in the true stress/strain curve used in the analysis.

I3.3 stress/strain properties

For stress/strain properties the following applies:

(a) Non-linear FEA uses the true stress true strain properties of the material.

(b) The following relationships may be used to convert engineering stress (σ) and engineering strain (ε) to true stress and true strain. These relationships are valid up to but not beyond the onset of necking at the maximum value of engineering stress (R_m).

$$\varepsilon_t = \ln(1+\varepsilon) \quad \sigma_t = \sigma(1+\varepsilon) \qquad \text{. . . I3.3(1)}$$

where ε_t is true strain and σ_t is true stress

(c) In those cases where the actual strengths of the material being used exceed the specified minimums (R_e and R_m) it is permissible to use a stress/strain relationship having—

(i) the average of the actual and minimum specified yield strengths;

(ii) the average of the actual and minimum specified tensile strengths; and

(iii) the average of the actual and minimum specified elongations.

(d) For those classes of vessel having a weld efficiency less than 1, the engineering stress/strain relationship shall be scaled down in proportion to the weld efficiency (see Note 1 of Table I1).

(e) If the stress/strain curve for the material is not available it is permissible to—

(i) assume elastic perfect plastic material;

(ii) assume an elastic linear true strain hardening relationship: or

(iii) approximate a true stress/strain curve from the specified minimum strengths of the material as follows (see Note 2 of Table I1).

For a given true stress, σ_τ, the corresponding true strain, ε_τ, is given by the following:

$$\varepsilon_t = \frac{\sigma_t}{E} + \ln(1+\varepsilon_\gamma)\left(\frac{\sigma_t}{R_{p0.2}(1+\varepsilon_\gamma)}\right)^{\frac{1}{m_1}} \frac{(1-H)}{2} + \frac{m_2}{e}\left(\frac{\sigma_t}{R_m}\right)^{\frac{1}{m_2}} \frac{(1+H)}{2} \qquad \text{. . . I3.3(2)}$$

A2

where

E = Young's modulus at the temperature of interest

e = the natural logarithm base 2.71828…

$$H = \tanh\left(\frac{2[\sigma_t - R_{p0.2} - K(R_m - R_{p0.2})]}{K(R_m - R_{p0.2})}\right) \qquad \text{. . . I3.3(3)}$$

$$K = 1.5R^{1.5} - 0.5R^{2.5} - R^{3.5} \qquad \text{. . . I3.3(4)}$$

$$m_1 = \frac{\ln(R) + \varepsilon_p - \varepsilon_\gamma}{\ln\left(\frac{\ln(1+\varepsilon_p)}{\ln(1+\varepsilon_\gamma)}\right)} \qquad \text{. . . I3.3(5)}$$

m_2 = curve fitting exponent from Table I1

$$R = \frac{R_{p0.2}}{R_m} \qquad \text{. . . I3.3(6)}$$

ε_τ = true strain

ε_p = curve-fitting parameter from Table I1

ε_γ = 0.002 (for 0.2% offset strain)

σ_τ = true stress

R_m = engineering ultimate tensile strength at the temperature of interest

$R_{p0.2}$= engineering proof strength at the 0.2% offset strain at the temperature of interest

The 1% proof strength properties $R_{p1.0}$ and $\varepsilon_\gamma = 0.01$ may be substituted for the 0.2% proof strength properties $R_{p1.2}$ and $\varepsilon_\gamma = 0.002$.

The development of the stress/strain curve should be limited to the value of true ultimate tensile stress ($R_{m,t}$) at true ultimate tensile strain, where

$R_{m,t}$ = true ultimate tensile stress at true ultimate tensile strain and

$$R_{m,t} = R_m e^{m_2} \qquad \text{. . . I3.3(7)}$$

The stress/strain curve beyond this point should be perfectly plastic (i.e. the true stress should be constant and equal to $R_{m,t}$).

A2

TABLE I1

STRESS/STRAIN CURVE-FITTING PARAMETERS

Material type	Temperature limit	m_2	ε_p
Ferritic steel	480°C	0.60(1.0 – R)	2.0×10^{-5}
Austenitic steel and nickel alloys	480°C	0.75(1.0 – R)	2.0×10^{-5}
Duplex stainless steel	480°C	0.70(0.95 – R)	2.0×10^{-5}
Precipitation hardenable nickel alloys	540°C	1.90(0.93 – R)	2.0×10^{-5}
Aluminium alloys	120°C	0.52(0.98 – R)	5.0×10^{-6}
Copper alloys	65°C	0.50(1.0 – R)	5.0×10^{-6}
Titanium and zirconium alloys	260°C	0.50(0.98 – R)	2.0×10^{-5}

NOTES:

1 To incorporate the reduction in strength implied by the weld efficiency, prior to generating the true stress/strain relationship, multiply the engineering stress and the strain at each point in the engineering stress/strain graph by the weld efficiency. It is necessary to multiply both stress and strain by the weld efficiency to preserve the gradients (such as Young's modulus) in the relationship.

2 These relationships are for use with the specified minimum strengths, not for strengths of the material in its ¼ hard ½ hard condition.

I4 BUCKLING

Non-linear analysis may be used for the buckling of vessels (e.g. knuckles) under internal pressure or vessels under external pressure.

Such analyses should take into consideration the following:

(a) Deviations from the ideal shape such as out of roundness and variations in thickness such as knuckle thinning and should be based on the actual shape and actual thickness less any corrosion allowance.

(b) Geometric non-linearity (changing shape with increasing load).

(c) Material non-linearity (non-linear stress/strain relationship above the yield strength).

(d) An appropriate factor of safety to determine the design pressure from the collapse pressure (on no occasion less than 2.0).

Extreme caution and considerable experience is required to evaluate FEA buckling results due to the highly variable sensitivity of structures to initial imperfections. The following safety factors are suggested where modelling includes thinning (e.g. typical of knuckles) but does not include out of roundness, and is based on the corroded thickness.

(i) 2.0 for knuckle radii on internally-pressurized dished ends.

(ii) 3.0 for cylinders under external pressure.

(iii) 14.0 for spheres or spherical components of dished ends.

I5 VIBRATIONS

Finite element analysis (linear or non-linear) may be used to determine resonant frequencies and associated stress and deflection distributions, excepting that if such stress distributions are to be used for fatigue analysis to Appendix M the relevant strains shall be determined and converted to quasi linear elastic stresses by multiplying by the appropriate Young's modulus.

APPENDIX J

WIND, SNOW AND SEISMIC LOADS

(Normative)

J1 GENERAL

This Appendix is intended to supplement AS/NZS 1170.0:2002, AS/NZS 1170.2:2002 and AS 1170.4—2007, providing data and requirements to allow application to pressure vessels. This Appendix is to be used for the design of pressure vessels when installed in Australia subject to any of the winds or seismic loads as defined in AS/NZS 1170.0:2002.

J2 IMPORTANCE LEVEL DETERMINATION

All pressure vessels shall be assigned a 'design working life' (as defined in AS/NZS 1170.0) of 25 years.

NOTE: This 'design working life' is intended only for the purpose of performing calculations according to the AS 1170 series, and is not necessarily related to expected service life of the vessel.

J3 ANNUAL PROBABILITY OF EXCEEDANCE

Pressure vessels shall be allocated an 'importance level' (as defined in AS/NZS 1170.0) according to Table J1, except for vessels located within or adjacent to a hazardous facility and for vessels where a detailed risk assessment has determined that an alternative hazard level is appropriate.

NOTE: Where it is identified that additional risks are present, a risk assessment might determine that a higher importance level is appropriate. The assessment of additional risks and the allocation of increased importance levels are an issue for agreement between the parties concerned.

TABLE J1

IMPORTANCE LEVEL FOR USE WITH AS/NZS 1170.0

Vessel hazard level to AS 4343	Importance level
A (high)	4
B (average)	3
C (low)	2
D (very low)	1
E (negligible)	1

Any pressure vessel located within or adjacent to a hazardous facility, which on failure could initiate a further event, shall be allocated an importance level 4 unless a detailed risk assessment determines that a lower importance level can safely be allocated.

A1 The probability of exceedance for a pressure vessel shall be determined according to Table 3.3 in AS/NZS 1170.0:2002.

J4 WIND DESIGN LOADS

J4.1 Windloads—Design wind speed

The regional wind speed (V_r) shall be determined from Table 3.1 of AS/NZS 1170.2:2002, according to the probability of exceedance.

For regions C and D, factors F_c and F_d (as defined in AS/NZS 1170.2:2002) may be reduced to a value that remains greater than 1.0, on the basis of a detailed study of recent meteorological data, the historical tracks and strengths of tropical cyclones and/or recorded wind speeds.

The regional wind speed (V_r) shall be converted to a permissible stress design speed (V_{des}) according to Equation J4.

$$V_{des} = 0.817 \times V_r \qquad \text{. . .J4}$$

J4.2 Design wind pressure

The design wind pressure (p) shall be calculated according to AS/NZS 1170.2:2002 Equation 2.4(1), using the rules set out in this Paragraph (J4.2).

Dynamic response factor (C_{dyn}) shall be taken as 1 for all vessels with a natural frequency greater than 1. Where vessel natural frequency is <1.0, refer to AS/NZS 1170.2:2002 Section 6.

The aerodynamic shape factor (C_{fig}) to be applied to the projected area of the vessel and/or any attachments, shall be according to Table J2.

TABLE J2

AERODYNAMIC SHAPE FACTOR

	Vessel type	Vessel h/d	C_{fig}
1	**Vertical vessels of circular cross-section (excluding stacks and chimneys)**		
1.1	Vessels (or partial sections) with negligible attachments outside the considered projected area—Essentially smooth surfaces	$h/d < 2$	0.65
		$2 < h/d < 7$	0.9
		$h/d > 7$	1.2 (Note 1)
1.2	Vessels with attached piping, ladders or platforms (e.g. typical process vessels)	$h/d < 7$	
	(a) Vessel—Essentially smooth surface		0.9
	(b) Vessel—Rough surface (e.g. corrugated cladding)		1.2
	(c) Attached structural components		1.2
	(d) Attached piping		1.0
1.3	Vessel with attached piping, ladders or platforms—with piping $h/d < 7$	$h/d > 7$	1.2
1.4	Vessels with attached piping, ladders or platforms—with piping $h/d > 7$	$h/d > 7$	
	(a) Vessel and other attached structural components		1.2
	(b) Attached piping		1.0
2	**Horizontal vessels**		
	(a) Vessel—Essentially smooth surface	—	1.0
	(b) Vessel—Rough surface		1.2
	(c) Attached structural components, supports		1.2
	(d) Attached piping		0.9

NOTES:

1 A value between 0.9 and 1.2 may be selected where the location is known, and where an engineering assessment determines that the wind conditions will allow a value in this range.

2 For further guidance on effects on platforms, ground gap etc, refer to PD5500 and enquiry cases. The attached piping and structural attachments may be calculated separately from the vessel, using a separate C_{fig}.

3 The factors above are intended to replace Note 2 to Table E3 in AS/NZS 1170.2:2002. Tables E3, E4 and E5 in AS/NZS 1170.2:2002 may be used for individual structural attachments, or a composite C_{fig} = 1.2 chosen as defined above, unless detailed assessment shows this value to be inappropriate.

J4.3 Design wind forces

Design wind forces shall be calculated using Equation 2.5(1) of AS/NZS 1170.2:2002, based on the wind pressures, V_{des}, C_{fig} and projected areas set out in this Appendix (J).

The wind direction multiplier (M_d) shall be 1.0 for Region A, and shall be 0.95 for Regions B, C and D .

J4.4 Site exposure multipliers

Site exposure multipliers (M_z) shall be as specified in Tables 4.1(A) and 4.1(B) of AS/NZS 1170.2:2002, as appropriate for the Region.

M_t and M_s shall be taken as 1.0 unless full details are provided of the installation of the pressure vessel.

J5 SEISMIC DESIGN LOADS

J5.1 General

Seismic design need not performed for vessels with a service weight of one tonne or less.

Seismic design shall be in accordance with AS 1170.4—2007 with modifications according to this Appendix (J).

NOTE: AS 1170.4—2007 has been primarily written for seismic design of multi-storey buildings and so does not easily lend itself to the determination of seismic forces for typical pressure vessel installations. This Appendix attempts to provide guidance for its application to pressure vessels however the designer should err on the side of conservatism where doubts exist as to its application.

Vessels classified as Importance Level 1 need not be designed for seismic loads, by agreement between the parties concerned.

Forces calculated using AS 1170.4 are aligned with the principles of ‘limit load analysis’. The forces determined per AS 1170.4 and modified by this Appendix shall be converted to ‘permissible stress forces’ (which is the basis of AS 1210 design) by dividing the calculated seismic force by 1.5.

J5.2 Design data

The following information shall be provided to the designer to allow seismic loads to be determined:

(a) Design working life (where greater than set out in Paragraph J2).

(b) Importance level (where greater than set out in Paragraph J3).

(c) Site sub-soil class.

(d) Installed height of vessel supports above grade and the type of vessel support, or a_c and R_c per Section 8 of AS 1170.4—2007.

(e) Floor acceleration at the base of the vessel when installed in an elevated structure and where seismic response is determined by the support structure design, weight and other relevant factors.

Where any of this information is unavailable, all assumptions shall be clearly documented.

Where floor accelerations are unavailable, the effect of installed height may be estimated by using Clause 8.3 of AS 1170.4—2007. Vessels that are at or within 1 m of grade may use $a_x = 1$

J5.3 Design practice

The procedure for seismic design shall follow Section 2 of AS 1170.4—2007, with modifications as set out in this Paragraph (J5.3).

This Paragraph (J5.3) refers to Sections, Clauses, Tables and Equations in AS 1170.4—2007, unless indicated otherwise.

Where information on the site sub-soil class has not been provided, the designer shall estimate the soil class according to Section 4, and document the assumptions made.

The 'earthquake design category' (EDC, defined in AS 1170.4—2007) shall be determined using Table 2.1.

NOTE: Vessels installed on typical concrete foundations up to 1m in height can be deemed to be single story composite installations for the purposes of this section. References to vessel height refer to the height of the centre of mass for horizontal vessels, and to the top of vertical vessels where $h/d > 2$.

For EDC I vessels (<12 m vessel height) use Equation 5.3. where W_i is the vessel weight.

For EDC I vessels (>12 m vessel height) and EDC II vessels (<15 m vessel height) use Equation 5.4 or Section 6. Use a floor factor K_s for a single storey structure from AS 1170.4—2007, Table 5.4 when the vessel is mounted at grade, and other factors from Table 6.5(B)

For EDC II horizontal vessels mounted within 1 m of grade, Section 6 may give overly conservative seismic forces. Section 8.3 may be alternatively used, but the minimum value shall be $0.1W_c$. For EDC II vessels (<15 m height) within a structure, Section 8 should be used.

For EDC II (>15 m vessel height) use Section 6, Equations 6.2(1) to 6.2(7) when mounted at grade. The natural period for the vessel will usually be determined from Equation 6.2(7), however if a more accurate analysis is available (e.g. FEA) then this may be used.

For EDC II (>15 m vessel height) within an elevated structure, Clause 8.2 should be used after a detailed assessment of the total structure's response has been completed. Any preliminary assessment of seismic forces prior to the completion of the detailed structural response after consideration of factors such as installed equipment shall be agreed between the parties concerned. Any assumptions should be validated by the structural designer prior to the completion of vessel fabrication.

NOTE: Typically the seismic weight at each vessel elevation will be equal to the design operating weight at that level—Equation 6.2(6) reduces to $W_i = \Sigma G_i$ as occupancy factors do not apply to vessels. Typically these forces will be calculated at each change in vessel diameter, section or thickness.

For EDC III use Clause 5.5 and Section 7. Only very tall vessels on soft soils or vessels of very high importance level typically fall into EDC III. In the event that a vessel falls under EDC III, the implication is that the vessel is dynamically sensitive and might need assessment beyond the typical simplified methods of Section 7. As Section 7 does not provide specific methods for conducting rigorous dynamic analysis of the equipment, it is recommended that competent personnel be used to conduct the modal or spectral analysis required to determine the response of the vessel to seismic actions.

The more rigorous methods of Section 7 are considered the minimum needed to ensure adequate seismic capacity as these vessels are likely to be more seismic sensitive and behave more dynamically than can be determined by the simple methods of Sections 5 and 6.

For horizontal vessels, it is acceptable to determine HD bolt and support structure design loads by applying the calculated seismic force at the vessel centre of gravity (COG) to determine base shear and overturning moments.

For vertical vessels, base shear and overturning moments can be determined by applying the calculated seismic acceleration at the vessel COG or by applying the seismic acceleration at the various section COG locations as appropriate. For combined stress calculations as per Clause 3.7.5 of this Standard (AS 1210), the vessel shall be broken down into discrete levels, and the weights and forces shall be calculated and applied at each level based on the COG location for the vessel portion above the section of interest.

For vessels installed within an elevated structure where the vessel operating weight is not greater than 15% of the total combined weight of the structure and equipment, the seismic forces on the vessel will be largely determined by the structure's dynamic response and not the vessel's response. Section 8 shall be used for the determination of seismic loads for this situation, as set out above.

For vessels installed in elevated structures where the vessel operating weight is greater than 15% of the total, a detailed analysis of the structure's response shall be carried out to determine the seismic acceleration at the vessel for use in Section 8.

J6 SNOW DESIGN LOADS

Where snow loads may be significant the design snow load shall be based on the probability of exceedance determined according to Paragraph J3.

J7 COMBINED LOADS

See Clause 3.2.3 for combination of loads.

APPENDIX K

LODMAT DATA FOR AUSTRALIA

(Informative)

FIGURE K1 LOWEST ONE DAY MEAN AMBIENT TEMPERATURE (LODMAT) (ISOTHERMS IN DEGREES CELSIUS)

APPENDIX L

COLD-STRETCHED AUSTENITIC STAINLESS STEEL VESSELS

(Normative)

L1 SCOPE AND APPLICATION OF APPENDIX

This Appendix specifies modified requirements for Class 1S and Class 2S vessels, referred to as cold-stretched vessels. These classes of pressure vessels are designed and fabricated from austenitic stainless steel and cold-stretched to enhance proof strength and permit higher design strengths.

Cold-stretched vessels shall comply with the requirements of this Standard, as modified by this Appendix.

L2 MATERIALS

L2.1 General

Materials shall comply with the requirements of Section 2 with the following additions.

L2.2 Material specifications

Material used for pressure parts or parts attached to pressure parts of cold-stretched vessels shall be restricted to one of the types of austenitic stainless steels listed in AS 1210 or an equivalent Standard, e.g. ISO 9328-7.

Only grades which are proven to be metallurgically stable at the intended service temperatures are permitted e.g. ASTM A240 Types 304, 304L, 304LN, 316, 316L, 316LN, 317L, 321, 347 for use between −196°C and +400°C. For other steel types or lower temperature, suitable type tests shall establish that the properties of parent metal, heat affected zone and weld metal are in accordance with this Appendix after 1000 hours at maximum and minimum design temperatures.

Steel shall be in the annealed (solution heat-treated) condition with an elongation of not less than 35% on a gauge length of 50 mm or equivalent except for cold-stretched or work hardened plate as specified in this Paragraph (L2.2).

Cold-stretched plate or work hardened (cold rolled) steel may be used provided that the following requirements are satisfied:

(a) The ratio $\frac{\text{actual 0.2 percent proof strength}}{\text{actual tensile strength}}$ shall not exceed 0.80.

(b) The maximum hardness shall not exceed that permitted by the original material specification.

(c) The elongation of the steel as supplied shall be not less than 30 percent on a gauge length of 50 mm or equivalent.

(d) The steel shall be shown, by suitable testing, to have a lateral expansion of at least 0.38 mm at the minimum operating temperature in the final operating condition. Suitable testing by the vessel manufacturer or steel maker shall be at least one set of Charpy tests on the steel type in both the maximum and minimum work hardened conditions, both welded and non-welded.

(e) The vessel manufacturer or steel maker shall document that the steel is suitable for the design and service conditions.

L2.3 Alternative material and component specifications

Clause 2.3.3 does not apply except when permitted by Clause 2.3.4.

L2.4 Material for high temperature service

Material selected for exposure to high temperature shall comply with Clause 2.7, except that the maximum service temperature for cold-stretched vessels shall be 400°C.

L2.5 Non-metallic materials

Non-metallic materials or components may be used for support and insulation, provided they are suitable for the intended service.

L3 DESIGN

L3.1 General

The requirements of Section 3 shall apply, except that Clauses 3.8, 3.13 to 3.17 and 3.27 to 3.32 shall not apply, and with the following additions.

L3.2 Design conditions

Specific reference should be made to Clause 3.2.5 for low-temperature service.

Particular care shall be taken to ensure that all openings and attachments (e.g. nozzles, pads and supports) are designed, located and fabricated so as to avoid discontinuities and sudden changes in stiffness or rigidity and to limit localized strain to 10% particularly in the circumferential direction. Calculations for the vessel support system shall be subjected to design verification. The calculations shall take account of all combinations of stresses due to restraint, discontinuities, thermal effects and vessel loadings including those under normal operating conditions and cold stretching.

Where practical, attachment welds should be arranged axially to allow free stretching.

In addition to the design requirements specified in this Appendix, it shall be shown by calculation or test that the permanent strain from cold stretching will not exceed 10% in areas of local strain concentration, except this is not required for dished ends complying with this Appendix.

The vessel shall meet the fatigue requirements of Clause 3.1.6 and Appendix M.

The vessel shall be constructed to Class 1 requirements except where this Appendix specifies other requirements.

L3.3 Design strengths

Clause 3.3 applies with the following modification:

The maximum allowable design tensile strength (f) shall be:

(a) For design temperatures up to and including 20°C, f is the lower of—

$R_e/1.5$; and

$$\frac{R_m}{2.5} \qquad \text{. . . L3.3}$$

where

R_e = specified minimum yield or proof stress (0.2% offset) at 20°C in the solution heat-treated condition, or in the specified supply condition, or it may be taken as the calculated stress actually developed at the cold-stretch pressure.

R_m = specified minimum tensile strength for the grade of material concerned at room temperature (tested in accordance with AS 1391 or equivalent), or

= 0.5 (Rm + RmACT) where agreed by the designer and the manufacturer. Actual tensile strength RmACT of each item of material used in the vessel, as recorded on the material maker's material test certificate. In this case the actual tensile strength is also limited to 125% of Rm. Also suitable tests are required to assess the actual strength of formed ends or other parts where fabrication methods may reduce tensile strength.

(b) For temperatures over 20°C and up to and including 400°C, the value from Item (a) shall be reduced by the amount in Table L3.3.

(c) For temperatures below 0°C, the provisions of Clause 3.33 may also be applied. For example, parts which can be assumed to be at low temperature when fully stressed, and are able to pass the hydro-test.

The factor 2.5 in (a) may be changed to 2.25 under the following conditions:

(i) f does not exceed 258 MPa.

(ii) Hydrostatic test pressure is not less than 1.5 × calculation pressure for all parts of the shell and ends.

(iii) Design temperature does not exceed 20°C and the design minimum temperature is below −50°C.

(iv) At least two pressure relief devices are fitted, one set at the design pressure (within allowable tolerance) and the other not greater than 1.21 times the design pressure.

(v) If the vessel is transportable, it also has external mechanical and thermal protection as specified in Paragraph L3.13.

TABLE L3.3

DESIGN TENSILE STRENGTH REDUCTION VALUES

Steel type (ASTM A240 or equivalent)	Design tensile strength, reduction values, MPa															
	Design temperature, °C															
	20	50	75	100	125	150	175	200	225	250	275	300	325	350	375	400
Standard types 304, 304L, 316, 316L	0	40	55	70	80	90	100	110	115	120	125	130	135	140	145	150
Ti or Nb—Alloyed types 321, 347	0	30	40	50	58	65	72	80	82	85	88	90	92	95	98	100
N—Alloyed types																
304LN, 316LN	0	60	80	100	112	125	138	150	158	165	172	180	188	195	202	210

L3.4 Welded and brazed joints

Clause 3.5 applies with the following additions and modifications:

(a) For location of joints, special reference should be made to Clause 3.5.1.3.

(b) For welded joints with backing strips as described in Clause 3.5.1.4.6 joggled joints shall not be used. Circumferential joints with retained backing strips may be used.

(c) For butt-welds between plates of unequal thickness (as in Clause 3.5.18), the thickness of plates in adjacent shell strakes and in ends of cold-stretched vessels may be varied provided—

(i) all plates in any one strake or in an end are the same nominal thickness;

(ii) the difference in thickness between plates in adjacent shell strakes do not exceed 20 percent of the thickness of the thinner plate; and

(iii) weld details between plates of different thickness comply with the requirements for Class 1H vessels in Clause 3.5.1.8.

NOTE: The thickness of ends and adjacent shell strakes is not limited by Item (b).

(d) Brazed and soldered joints shall not be used except for attached piping.

L3.5 Cylindrical and spherical shells subject to internal pressure and combined loadings

L3.5.1 *General*

Clause 3.7 applies with the following additions.

L3.5.2 *Vertical cylindrical vessels under combined loading*

For cold-stretched vessels, suitable provision shall be made to avoid axial buckling due to shells distorted unequally by stretching. See Paragraph L3.5.3 for design for hydrotesting in the horizontal position.

L3.5.3 *Horizontal cylindrical vessels under combined loading*

The design shall cater for any axial buckling during hydro stretching and for modified shell shape.

L3.6 Cylindrical and spherical shells subject to external pressure

L3.6.1 *General*

Clause 3.9 applies with the following modifications.

L3.6.2 *Notation*

The value of f shall be taken as that given in Clause 3.9.2.

L3.6.3 *Stiffening rings for cylindrical shells subject to external pressure*

For the form of stiffening rings in Clause 3.9.6.2, circumferential rings are permitted as stiffeners or supports of cold-stretched shells, provided the rings are complete so that no openings or notches are formed which cause localized strain in the shell or ring in excess of 10%. Any openings or notches in the rings shall be smooth. Such openings or notches should be located clear of joints in the ring. The joints in the ring itself shall be full penetration welds, subject to radiography before stretching and to close visual examination after stretching, where accessible. Toes of welds on the shell plate, of welds attaching the ring to shell, shall blend smoothly and be free of undercut. See Paragraph L3.12.

L3.7 Dished ends subject to internal pressure

Clause 3.12 applies with the addition that ends shall be spherical or ellipsoidal with an outside major to minor axis ratio not greater than 2:1.

L3.8 Openings and reinforcements

L3.8.1 *General*

Clause 3.18 applies with the following additions and modifications.

NOTE: Allowance should be made for possible distortion of the opening and nozzle, e.g. by use of reduced value of f.

L3.8.2 *Location of openings*

Any opening exceeding 325 mm diameter shall be located in the end of the vessel and shall be concentric with the vessel shell except where it is demonstrated that the local strain in the end does not exceed 10%.

NOTE: Designs should generally have all attachments located in the vessel ends unless impractical.

L3.8.3 *Unreinforced openings*

Unreinforced openings shall comply with Clause 3.18.6 using the requirements for Class 1H vessels.

Nozzles should be of the same type of material as the shell. Integral reinforcement of the opening by increased nozzle wall thickness is preferred in all cases. Doubling plates or pads (or reinforcing rings around the nozzle) should be avoided where practicable to prevent excessive local strain in the shell near the pad.

L3.8.4 *Reinforcement of single openings in shells and dished ends*

For cold-stretched vessels the design of openings and reinforcements (see Clause 3.18.7) shall be on the area replacement method (i.e. plastic design) assuming the shell and nozzle have an '*f*' value not in excess of that determined in Table A1.

L3.9 Connections and nozzles

Clause 3.19 applies with the addition that, in Clause 3.19.3.4, tell-tale holes are not required in internal doubling or wrapper plates where heat treatment is not required and avoidance of content contamination is necessary.

L3.10 Jacketed vessels

Clause 3.23 applies, except for partially jacketed cold-stretched vessels.

L3.11 Vessel supports

Clause 3.24 applies in conjunction with Paragraph L3.12.

L3.12 Attached structures and equipment

Clause 3.25 applies with the following additions to Clause 3.25.4:

Rings and similar structural parts attached to cold-stretched shells or ends shall comply with the following:

(a) Clause 3.9 for rings designed for prevention of buckling.

(b) Clause 3.24.4, AS 3990, or other appropriate reference for supports using design strength (*f*) as in this Standard at a design temperature equal to—

  (i) at least 20°C, to ensure adequacy at the combined pressure and loading during the hydro or other pressure test; and

  (ii) the highest temperature of liquid adjacent to the vessel part under coincident service pressure and load conditions, e.g. for cryogenic transportable vessels, supports and internal stiffeners may be designed for the required *G* loading, with *f* at the boiling point of the contained gas at the safety valve pressure setting.

(c) As an alternative to Item (b), comply with AS 4100 but with yield strength equal to the 1% proof strength of the steel at the temperatures specified in Item (b).

(d) Calculations for support rings shall include determination of transverse bending stress in the ring flange.

(e) Welds attaching rings to the shell shall be continuous on both sides of the web with a minimum total throat thickness equal to the web thickness. See Paragraph L3.9.

L3.13 Transportable vessels

Clause 3.26 applies with the following modifications:

(a) In Clause 3.26.3.1, transportable cold-stretched vessels for the carriage of products by road, rail or sea shall be Class 1S construction in place of Class 1, and shall be Class 2S in place of Classes 2A and 2B.

(b) The design shall be assessed and comply with the fatigue requirements of Paragraph L3.2 taking into account actual shape of welds permitted, local distortion, location of attachments and the additional transport loads. This shall also apply to the supports.

(c) Before use with lethal or very harmful fluids (see AS 4343), the design shall be proven by operation of a vessel fully representative of the design and construction, for a service period of at least one year using fluid other than lethal or very harmful; or by accelerated fatigue testing a full size vessel under equivalent loading and cycling.

(d) For ISO tank container type pressure vessels, the total assembly of the frame and vessel shall meet the requirements of the relevant authority for the particular mode of transport to be carried by sea and the relevant rail authority for containers to be carried by rail.

(e) For vessels covered by Item (d), the design strength (f) used in the design of shells and ends shall comply with the IMDG code.

(f) The type of material, its design strength, thickness and any insulation shall be sufficient to give a period of at least 15 minutes from the time of fire engulfment to rupture of the shell. The rupture of the shell shall not result in fragmentation. Supporting calculations or tests shall be made except for the Items (f) where jacket metal is of steel.

Calculations shall be based on any insulation being intact except for loss of any vacuum, and assuming uninsulated gas backed surfaces reaching 650°C, the pressure equalling the design pressure, and reliable short-term mean creep rupture data of the steel at 650°C.

A2 (g) Transportable non-vacuum, cold-stretched vessels shall have external protection against impact at least equivalent to—

(i) 2 mm metal jacket with 100 mm powder or fibre insulation;

(ii) 2.5 mm metal jacket with multi-layer (super) insulation; or

(iii) 1 mm metal jacket with 100 mm rigid fire retardant foam.

For lethal or very harmful (toxic or flammable) contents, the combined thickness of the metal jacket and the vessel wall shall be at least 9 mm. The minimum thickness of the metal jacket shall be 2 mm.

For harmful or non-harmful contents, the combined thickness of the metal jacket and vessel wall shall be at least 7 mm.

NOTE: Insulation for impact protection may also be used for fire protection.

L3.14 Design against impact or collision

Vessels with a credible risk of being subjected to impact or collision shall be designed to minimize damage and loss of containment by:

(a) Increased thickness or material strength.

(b) Use of material with value of $\left(\dfrac{R_e + R_m}{200}\right) \times A_5 \leq 100$, where A_5 is as defined in Paragraph A2.

(c) Use of Class 1 or lighter construction.

(d) Keeping attachments, fittings and parts within the outline of the vessel and for such parts to be shaped to minimize direct impact or damage to others (e.g. avoid sharp corners on skid plates).

(e) Providing protection e.g. insulated jacket or equivalent, or by guards etc. as required in Clause 3.26 or by safeguarding (crash barriers etc.).

In such cases vessel shell thickness shall also comply with Clause 3.26.

L3.15 Vessels with increased design strength at low temperature

Clause 3.33 applies with the provision that alternative reliable and conservative strength values for cold-stretched steel may be used. Such design strength values shall be fully justified at the design verification.

L4 MANUFACTURE

L4.1 General

The manufacture of cold stretched vessels shall comply with Section 4 with the following modifications.

L4.2 Welded construction—General welding requirements

Clause 4.2.1 applies with the following modifications:

(a) Butt weld reinforcement limits to this Standard and AS 4458 apply, but lower values are recommended for longitudinal type welds.

(b) Dressing of welds is only required where necessary to meet these limits or the shape required by fatigue design.

L4.3 Brazed construction

In Clause 4.5, brazing and soldering shall not be used on main shells and ends of cold-stretched vessels.

L5 TESTING AND QUALIFICATION

L5.1 General

The requirements of Section 5 shall apply, with the following additions and modifications to particular Clauses.

L5.2 Non-destructive examination

Clause 5.3 applies with the following additions:

(a) Each vessel after cold-stretching shall be examined at accessible surfaces at the circumferential rings (and other areas of potential high localized strain) to ensure local strain has not exceeded the 10% limit.

(b) Radiographic examination shall be done before the hydrostatic test. In Class 1S vessels, longitudinal and circumferential type welds shall have 100% radiographic examination. In Class 2S vessels longitudinal type welds shall have 100% radiographic examination and circumferential type welds shall have at least 10% radiographic examination. All tee joints shall be radiographed.

(c) Penetrant examination shall be done after the hydrostatic test at areas which are accessible and the local strain may exceed 5%.

L5.3 Hydrostatic tests

Clause 5.10 applies with the following addition and modification to Clause 5.10.2.1:

The vessel shall be hydrostatically tested and cold-stretched at the higher pressure calculated in accordance with Equations L5.3(1) and L5.3(2).

$$P_{\mathrm{h}} = 1.5\,P \times \frac{f_{\mathrm{h}}}{f} \qquad \text{. . . L5.3(1)}$$

A1

$$P_{\mathrm{h}} = 1.5 \times \frac{f_{\mathrm{h}}}{f}(P + h\,g\,\rho \times 10^{-6}) - (h_{\mathrm{h}}\,g\,\rho_{\mathrm{h}} \times 10^{-6}) \qquad \text{. . . L5.3(2)}$$

where

P_{h} = hydrostatic test pressure at the top of the vessel, in megapascals

P = design pressure of the vessel at the top of the vessel, in megapascals

f_{h}/f = lowest ratio (for the materials of which the vessel is constructed) of—

$$\frac{\text{design strength at test temperature, MPa}}{\text{design strength at design temperature, MPa}} 84$$

(Values to be taken from Table L3.3 or as used in the design.)

h = maximum head of liquid in service, in metres

h_{h} = actual head of test liquid, in metres

ρ = density of liquid in service, in kilograms per cubic metre

ρ_{h} = density of test liquid, in kilograms per cubic metre

g = 9.81, in metres per second per second

The test pressure shall be maintained until the measured strain rate does not exceed 0.1% per hour.

The total circumferential strain shall be determined during pressuring and depressuring by measuring the general increase in circumference of at least one strake of each nominal thickness at positions which are judged to have minimum restraint.

NOTE: In a vessel which does not have circumferential stiffening, the above mentioned locations will normally be at the mid-length of the shell strakes having the lowest value of plate thickness multiplied by proof strength (from the steel makers certificates). To facilitate measurement this strake may be located at the bottom of large vessels tested in the vertical position.

The total permanent circumferential strain based on the general increase in circumference shall not exceed 5 percent.

In the event the required test pressure is not reached at 5.2% total strain, the vessel shall be modified, rejected or derated to a design pressure equal to $\frac{1}{1.5}$ times the maximum test pressure reached at 5.2% total strain.

L5.4 Special examinations and tests

Clause 5.17 applies with the addition that a full scale test shall be made to ensure the transverse strain at the toes of fillet welds attaching a circumferential ring to the shell are within strain limits. Measurements to check this strain shall be taken after completion of the cold-stretch/hydrostatic test at a point located on the vessel wall showing maximum curvature from the original shape. This location may be determined by visual inspection and shall account for any welding distortion that may be present before cold stretching is carried out. This test may be made on one completed vessel with the highest total circumferential strain or may be made on a representative test specimen made of the same materials and thickness. Where specified, an agreed thermal shock test may also be made.

L6 MARKING

L6.1 General

Marking requirements of Section 7 shall apply, with the following additions and modifications.

L6.2 Marking required

In addition to the marking required by Clause 7.1, each cold-stretched vessel shall be marked with the maximum permanent strain actually measured, e.g. '1.2% max. cold stretch'.

L6.3 Reports

The Manufacturers Data Report (see AS 4458) shall show the marking specified in Paragraph L6.2 and the location of this strain.

A record of the pressurizing sequence including measured pressures and circumferential measurements versus time and strain rate shall be kept by the manufacturer.

APPENDIX M

DESIGN AGAINST FATIGUE

(Normative)

M1 GENERAL

During the operation of pressure vessels, important parts may be subjected to cyclic or repeated stresses. Such stresses may be caused by—

(a) applications or fluctuations of pressure;

(b) periodic temperature transients (see M11);

(c) vibrations (forced and resonant) (see M12); or

(d) variations in external loads.

Under these conditions the failure mode (fatigue cracking) will occur during the operational life if the fatigue strength of the material used in any part of the pressure vessel is exceeded for the particular number of repeated stress cycles. The potential for fatigue cracking is reduced when the magnitude of cyclic stress, or the number of cycles, or both, is reduced.

This Appendix gives requirements and recommendations to avoid fatigue cracking and failure. This Appendix is based on design and manufacture complying with all of the requirements of this Standard.

M2 DESIGN FEATURES

Where vessels are subject to many cycles of stress, consideration shall be given in the design to reducing the risk of fatigue by giving particular attention to the possible deleterious effects of the following design features:

(a) Non-integral constructions, such as the use of pad type reinforcements or fillet welded attachments, as opposed to integral construction, particularly if cyclic temperature changes are present.

(b) Pipe threaded connections, particularly for pipe outside diameters in excess of 65 mm.

(c) Studded flange connections, e.g. Figure 3.19.6.

(d) Partial penetration welds.

(e) Major changes in thickness or direction between adjacent members.

(f) Welds with poor toe shape.

(g) Rough surfaces near the junction of the bore of openings and the vessel ID.

M3 DEFINITIONS

For the purposes of this Appendix, the definitions below apply.

A1 (a) *Cut-off limit* (applicable to steels only)—the largest variable stress range that does not require consideration when carrying out cumulative damage calculations (see Figure M1, dashed line or the value of S_r at 10^8 cycles).

(b) *Design life*—period over which the vessel or element is required to perform its function without repair.

(c) *Discontinuity*—shape, or material change that affects the stress distribution.

(d) *Endurance limit*—cyclic stress range below which, in the absence of any previous loading, no fatigue damaged is assumed to occur under constant amplitude loading.

(e) *Fatigue*—damage caused by repeated fluctuations of stress leading to gradual cracking of a vessel element.

(f) *Fatigue strength* (S_r)—the allowable stress range for the number of stress cycles, N. (See Figure M1.)

(g) *Geometric stress*—the stress at the location of a weld toe that includes the primary plus secondary stress components but excludes the peak stress component.

NOTE: This stress includes the primary plus secondary stresses associated with gross structural discontinuities (e.g. nozzle connections, cone/cylinder intersections, vessel/end junctions, thickness change, deviations from design shape, presence of an attachment). The geometric stress is obtained by extrapolating the trendline of primary plus secondary stresses from locations between approximately 3 t to 0.4 t (where t is the plate thickness) from the weld toe (see Figure M5). This excludes the peak stress component at the weld toe which is insignificant at distances greater than 0.4 t.

(h) *Gross structural discontinuity*—structural discontinuity that affects the stress distribution across the entire wall thickness. (See Paragraph H2.)

(i) *Local structural discontinuity*—discontinuity that affects the stress distribution locally, across a fraction of the wall thickness (i.e. causes a stress concentration) defects such as flaws, imperfections or indentations, cracks, scratches, gouges, corrosion pits, lack of penetration, slag inclusions, cold laps, porosity, undercut and sharp corners. See 'notch' in Paragraph M10(a).

NOTE: Cracks or sharp changes of direction are singularities, the most extreme form of local structural discontinuity.

(j) *Miner's summation*—cumulative damage calculation based on the Palmgren-Miner summation or equivalent.

(k) *S_r–N curves*—curves defining the limiting relationship between the permissible number of stress cycles (N) and stress range (S_r). See Figure M1.

(l) *Stress concentration*—a source of peak stress. (See Paragraph H2.)

A1 (m) *Stress cycle*—one cycle of Tresca stress as defined by stress cycle counting. This is established from the changes in the component stresses between extremes of the cycle at the point being considered.

NOTE: It is not valid to determine a Tresca stress intensity range by taking the difference between Tresca stress calculated at the extremes of the cycle. For example, given the following case where principal stress directions do not change and the three principal stresses in directions a, b and c cycle between the extremes:

$\sigma_a = +100$ MPa, $\sigma_b = 0$ MPa, $\sigma_c = -200$ MPa

and

$\sigma_a = -200$ MPa, $\sigma_b = 0$ MPa, $\sigma_c = +100$ MPa

that is the principal stresses range through

$\Delta\sigma_a = -300$ MPa, $\Delta\sigma_b = 0$ MPa, $\Delta\sigma_c = +300$ MPa

and accordingly, the range in Tresca stress is 300 − (−300) = 600 MPa.

As the Tresca stress is exactly the same at both extremes (i.e. 300 MPa) such that taking the difference between the Tresca stresses at the extremes would give an erroneous zero result for the Tresca stress range. An identical observation may be made with respect to calculation of the von Mises stress intensity range.

(n) *Stress cycle counting method*—any rational method used to identify individual stress cycles from the stress history. The preferred method is the Rainfall or Reservoir method.

(o) *Stress range*—algebraic difference between two extremes of stress, i.e. 2 × amplitude.

(p) *Stress spectrum*—histogram of the stress cycles produced by a nominal loading event.

M4 FATIGUE LIFE

M4.1 Need for fatigue analysis

Fatigue analysis is:

(a) Not required for Class 1, 2 and 3 vessels, but should be considered for high hazard vessels of Hazard Levels A and B with more than the number of cycles of full pressure range, or equivalent, given in Table 1.6 (Item 3.5, Fatigue Assessment).

(b) Required for Class 1H, 2H, 1S or 2S vessels, except where—

(i) the number of cycles of full pressure range ($\Delta P = P$) does not exceed 500 and there are no significant cyclic stresses associated with temperature fluctuations or non-pressure loads;

(ii) for a pressure range ΔP lower than P, the number of cycles does not exceed $500\left(\dfrac{P}{\Delta P}\right)^3$ and there are no significant cyclic stresses associated with temperature fluctuations or non-pressure loads; and

(iii) the stress range determined using the Equation M4.1 is less than the stress range determined from the 'relevant curve' in Figure M1 at the relevant number of cycles.

$$S_r = \frac{e}{\eta}\frac{200000}{E}\left\{\frac{f \cdot \Delta P}{P} + E\alpha\Delta T\right\} \quad \text{. . . M4.1}$$

where

E = Young's Modulus of elasticity at the highest temperature during the cycle at the location giving the maximum value of the product $E\alpha\Delta T$ (see Table B3), in megapascals

e = the maximum applicable stress concentration factor from the following list:

= 2.5 for vessels with dressed, full penetration welds

= 4.0 for vessels with undressed welds and integral reinforcing

= 6.0 for vessels that contain unreinforced openings

= 10.0 for vessels with non-integral reinforcing or fillet welded attachments

f = material design tensile strength from Table B1, in megapascals

ΔP = operating pressure range, in megapascals

NOTE: In the case of multiple operating cycles, Equation M4.1 may be used in combination with Equations M4.2 and M4.4.

P = design pressure, in megapascals

NOTE: At the discretion of the designer, the design pressure may be inflated to improve fatigue life.

S_r = Tresca stress range from the relevant fatigue curve (see Figure M1)

ΔT = temperature range at the location giving the maximum value of the product, $E\alpha\Delta T$

α = coefficient of thermal expansion for the location giving the maximum value of the product, $E\alpha\Delta T$, in mm/mm°C

η = weld efficiency

NOTES:

1 It is conservatively assumed for the purpose of this Clause that the component of stress intensity range for a given load cycle, associated with the fluctuation in temperature, results from fully retrained thermal expansion (or contraction) and is fully additive to the component of stress amplitude associated with the pressure fluctuation during that cycle, either component may be zero for a given load cycle.

2 The values of ΔP and ΔT are always positive, for the purposes of this equation.

3 Equation M4.1 is applicable for all vessel classes.

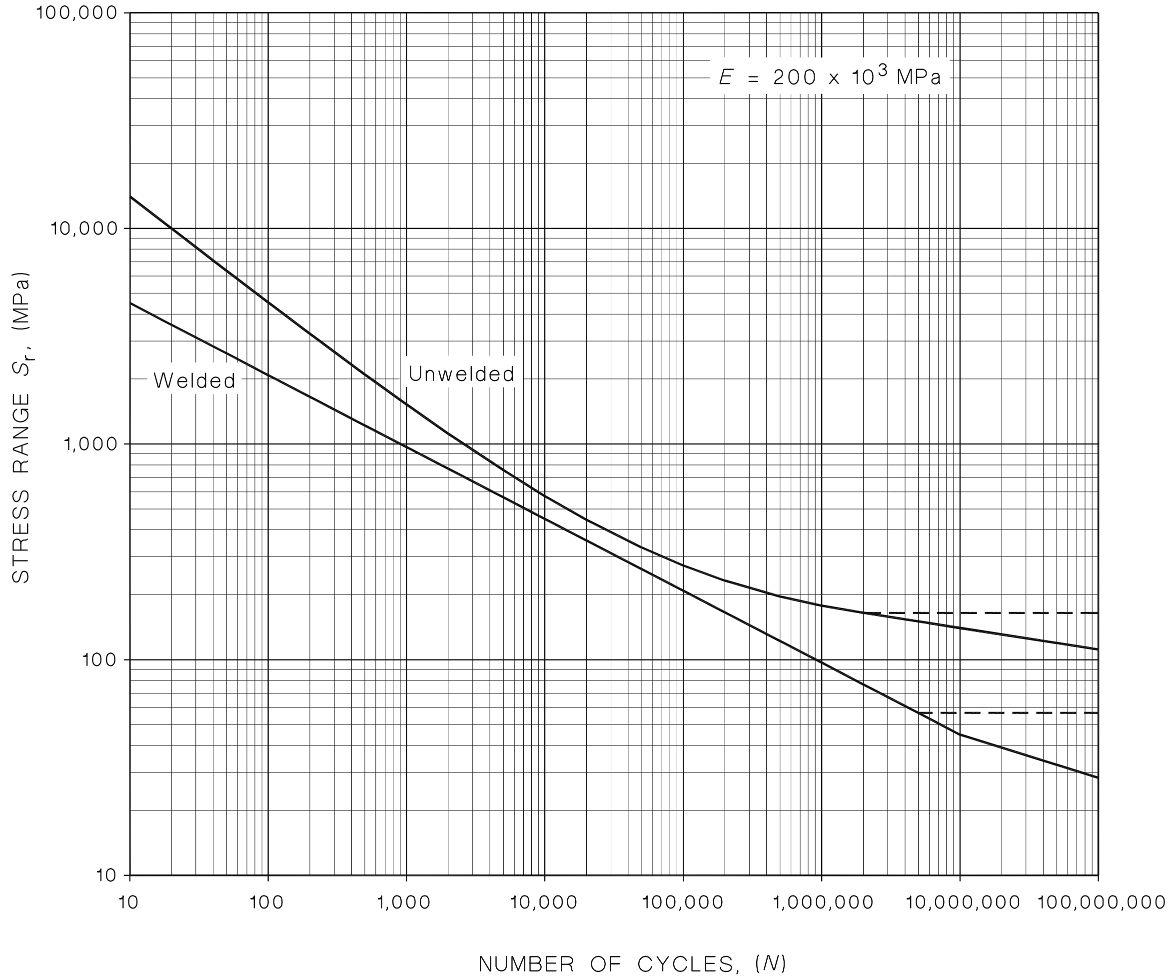


FIGURE M1 S_r–N DESIGN FATIGUE CURVES FOR METALS OTHER THAN HIGH STRENGTH STEEL BOLTING FOR TEMPERATURES <375°C

M4.2 Equations for fatigue design curves in Figure M1

M4.2.1 *Stress range for variable amplitude loading—full curves in Figure M1*

(a) *Unwelded*:

$$N = \left(\frac{44022}{S_r - 134}\right)^2 \text{ for } 10 \le N \le 2 \times 10^6 \qquad \text{. . . M4.2(1)}$$

or

$$N = \frac{A}{S_r{}^m} \text{ for } 2 \times 10^6 < N \le 10^8 \qquad \text{. . . M4.2(2)}$$

where

$A = 3.9 \times 10^{28}$

$m = 10$

(b) *Welded*:

$$N = \frac{A}{S_r{}^m} \text{ for } 10 \le \mathrm{N} \le 10^7 \qquad \text{. . . M4.2(3)}$$

where

$A = 9.1135 \times 10^{11}$

$m = 3$

or

$$N = \frac{A}{S_r{}^m} \text{ for } 10^7 < N \le 10^8 \qquad \text{. . . M4.2(4)}$$

where

$A = 1.8376 \times 10^{15}$

$m = 5$

A1 **M4.2.2** *Stress range for steels with constant amplitude loads—that is, dashed curves on Figure M1*

(a) *Unwelded*:

$$S_r = 165 \text{ for } N > 2 \times 10^6 \qquad \text{. . . M4.2(5)}$$

(b) *Welded*:

$$S_r = 57 \text{ for } N > 5 \times 10^6 \qquad \text{. . . M4.2(6)}$$

M4.3 Basis of S_r–N curves

The fatigue curves given in Figure M1 are intended to be sufficiently conservative such that they are reasonable to use regardless of the metal, mean stress and statistical scatter in the source data. They are based on (mean − 2σ limits) of a wide range of fatigue tests in dry air and with a value of:

$$R\left(=\frac{\text{minimum stress}}{\text{maximum stress}}\right) \text{ of approximately 0.5, which allows for residual stress.}$$

The welded S_r–N curve allows primarily for metallurgical damage i.e. micro-cracks which accelerate macro-crack initiation and the small highly localized stress concentration associated with the weld toe radius.

M4.4 Stress cycles of different magnitude

If there is more than one magnitude of stress cycle then the combined effect of these cycles shall be determined by calculating the Cumulative Damage Factor (U) by the following equation:

For a vessel having load cycles 1, 2, 3 etc.

$$U = n_1/N_1 + n_2/N_2 + n_3/N_3 + \ldots \text{ etc.} \qquad \ldots \text{M4.4}$$

where

n_1 = actual number of cycles of load cycle 1 applied to the vessel, similarly for n_2, n_3 etc.

N_1 = maximum permissible number of cycles of load cycle 1 where it is the only load cycle applied to the vessel, as determined from Figure M1, similarly for N_2, N_3 etc.

N_1 is determined from Figure M1 using the stress range S_{r1} where S_{r1} is the stress range associated with load cycle 1. N_2, N_3 etc. are similarly determined.

The Cumulative Damage Factor, U, shall not exceed 1.

The operating pressure ranges (ΔP_1, ΔP_2, etc) and the number of cycles (n_1, n_2, etc) shall be obtained by—

(a) simple counting method; or

(b) rainfall or reservoir cycle counting method.

Cycles of $\Delta P \leq 0.05P$ may be ignored.

An example of using the rainfall or reservoir cycle counting method is shown below. Consider the example pressure cycle history, shown in Figure M2.

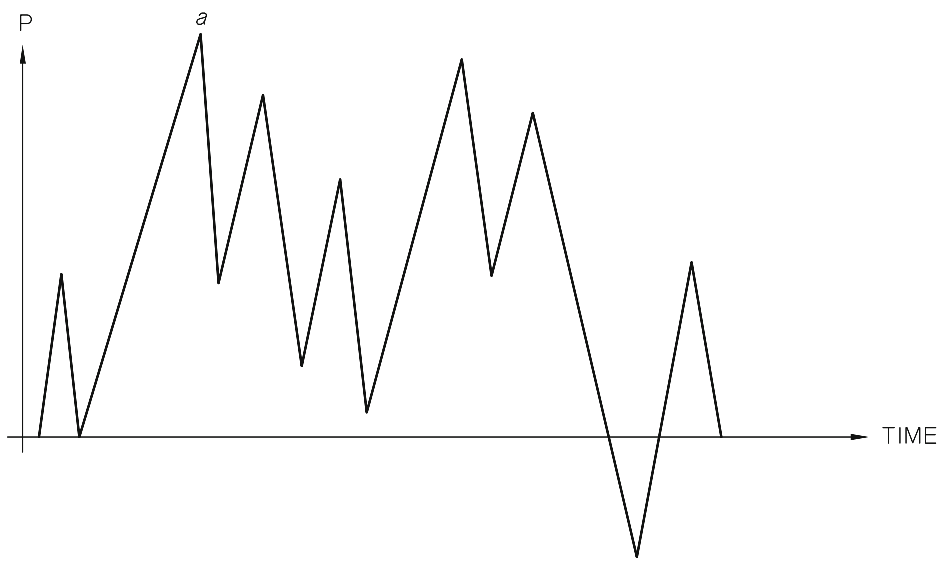


FIGURE M2 RAINFALL OR RESERVOIR COUNTING METHOD

Firstly, determine the highest pressure in the pressure time history (in this case Point a). Remove the cycles up to this time and append them onto the end of the history, as shown in Figure M3.

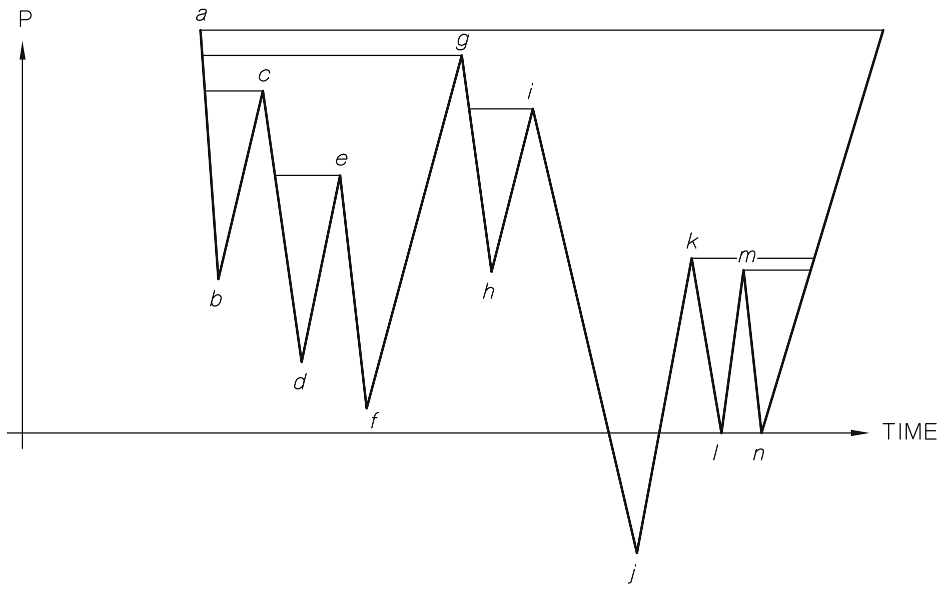


FIGURE M3 ‘DRAIN’ POINTS FOR EXAMPLE PRESSURE CYCLE HISTORY

Consider the resulting shape to be a trough filled with water. Take the deepest depth of water as a pressure cycle (*aj* in this case).

Drain the trough from the deepest point (j in this case) and take each of the maximum depths of the remaining troughs as further pressure cycles (*gf*, *ih* and *kl* in this case).

Continue to drain each remaining trough taking the deepest depths as pressure cycles until there is no remaining water, giving in this case the pressure cycle history:

aj, *gf*, *ih*, *kl*, *cb*, *ed*, and *mn*,

Each part of this history is then designated as load cycle 1, 2, etc, and ΔP and n identified for each load cycle, for use in equation M4.4.

M5 METHODS OF ANALYSIS

Fatigue analysis may be performed by—

(a) simple assessment, Paragraph M4.1;

(b) detailed analysis, Paragraph M7;

(c) experimental analysis, Paragraph M9;

(d) fracture mechanics, Paragraph M9;

(e) prior successful experience with comparable or more severe service stresses and conditions; or

(f) a combination of the above methods.

M6 ADJUSTMENTS

M6.1 General

Adjustments to the calculated stress ranges and/or the fatigue strength of the material shall be carried out as follows.

M6.2 High triaxiality

The stress range for use with the fatigue curves given in Figure M1 is the range in peak or geometric Tresca stress intensity. An exception to this are those cases of high triaxiality, where all three principal stresses are of the same sign (i.e. all three principal stresses at the point under consideration are either tensile or compressive). In such cases the range in each of the three principal stresses shall be determined and the maximum range of these three used with the relevant fatigue curve in Figure M1. Those cases in which the range in maximum principal stress exceeds the range in Tresca stress shall be considered as cases of high triaxiality.

M6.3 Out of phase stresses

An additional fatigue strength reduction factor of 2 (applied by dividing S_r taken from Figure M1 by 2) shall be applied where the component stresses in a multiaxially stressed location fluctuate out of phase.

M6.4 Young's Modulus adjustment

Fatigue damage is essentially a strain related phenomenon and accordingly a given number of cycles of a particular stress range applied to a material with Young's Modulus of 200 GPa will result in approximately the same fatigue damage as half that stress range for the same number of cycles applied to a metal with Young's Modulus of 100 GPa.

The fatigue curves given in Figure M1 (and the associated data in Paragraph M4.2) are for a metal having a Young's modulus of 200 MPa. Stress ranges for metals having different Young's Moduli shall be appropriately adjusted prior to determining N values from Figure M1. For example if a peak or geometric stress range of 80 MPa is determined for a 6061 aluminium alloy at 150°C then the appropriate stress range to be taken to Figure M1 is $80 \times 200/63$ i.e. 254 MPa, given that this alloy has a Young's Modulus of 63 GPa at 150°C (see Table B3).

M6.5 Corrosion fatigue

Corrosion conditions are detrimental to the endurance of carbon steels, carbon-manganese steels, ferritic alloy steels and some other metals. Fatigue cracks may occur under such conditions at low levels of fluctuation of applied stress. Since the tensile strength of a steel has little or no effect upon the fatigue strength under corrosive conditions, the use of high strength steels in severe corrosion fatigue service will offer no advantage unless the surface is effectively protected from the corrosive medium.

Where corrosion fatigue is expected, it is desirable to minimize the range of cyclic stresses or use lower stress levels, limit the number of cycles or control corrosion and carry out inspection at sufficiently frequent intervals to establish the pattern of behaviour.

Where the environment is other than reasonably dry air, S_R values in Figure M1 should be adjusted in accordance with authoritative data and with agreement of parties concerned. For example, S_R values taken from Figure M1 may be:

(a) Increased by 10% for a high level vacuum or inert dry fluids.

(b) Taken as in Figure M1 for air and most reasonably dry non-corrosive gases.

(c) Decreased by up to 50% in sea water or corrosive environment.

(d) Decreased by up to 20% for high temperature scaling.

(e) Decreased at $N > 2 \times 10^6$ cycles in recognition there is no endurance limit under corrosive conditions.

NOTE: See EN 13445 for design limits to avoid repeated fracturing of magnetite layer in steam and hot water service.

M6.6 In-service inspection

Fatigue life predictions are intended to be supplemented by in-service inspections particularly toward the end of the design life. Such inspections can be expedited by the recording and record keeping of fatigue-prone locations. These are generally few in number and highly localized on any one vessel. Such ‘hot spot’ stresses should be determined and identified at the design and stress analysis stage.

Where such inspections are not feasible, the S_r values taken from Figure M1 shall be multiplied by 0.7.

M6.7 Non-destructive examination

Where the potential source of fatigue cracking is not examined by visual examination and non destructive examination, the calculated stress range shall be multiplied by 1.25.

A2 **M6.8 Enhancement of fatigue performance of weld toes**

M6.8.1 *General*

The fatigue performance of a weld toe can be enhanced by any one of the following methods:

(a) Hammer peening.

(b) Ultrasonic impact treatment.

(c) TIG toe dressing.

(d) Underflushing by toe grinding.

M6.8.2 *Beneficial effect*

Incorporation of the beneficial effect of such weld toe enhancement is achieved by multiplying the calculated geometric stress range by the following factor, F_{wt}:

Enhancement method	F_{wt}	Applicable stress range
Hammer peening	0.69	$S_r \leq R_{eT}$
Ultrasonic impact treatment	0.72	$S_r \leq R_{eT}$
TIG toe dressing	0.77	All
Toe grinding, no underflushing	0.79	All
Toe grinding with underflushing	0.70	All

Where, S_r is the stress range on the welded curve (Figure M1) being the maximum unenhanced calculated geometric stress range being considered (including that associated with hydrostatic testing). The enhanced geometric stress range ($F_{wt} \times S_r$) is then used to determine the permissible number of cycles (N) from the welded curve (Figure M1). Only one enhancement factor F_{wt} shall be chosen from those listed in the table for any given geometric stress location.

R_{eT} is the yield strength at the operating temperature of the parent metal adjacent to the weld toe considered.

M6.8.3 *Underflushing*

Underflushing of weld toes shall satisfy the following requirements:

(a) Part thickness shall be at least 10 mm thick.

(b) The underflushing shall be ground or machined to a depth of between 0.5 and 1.0 mm (see Figure M4) to effectively remove undercut and/or microcracking, and shall have any resulting grinding/machining marks both—

 (i) minimized as far as possible; and

 (ii) running transverse to the weld toe direction.

(c) To prevent unacceptable loss of section strength underflushing shall not exceed 5% of the section thickness.

A2 (d) The dressed area shall be examined using magnetic particle or dye penetrant examination in compliance with AS 4037.

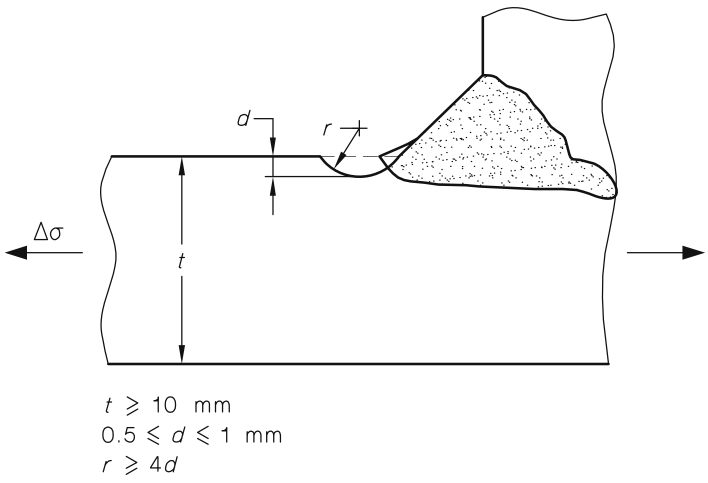


FIGURE M4 WELD TOE DRESSING

M6.9 Thickness

For a part containing surface welds, where the thickness of the part (t_p) is over 25 mm, multiply S_r by $\left(\frac{25}{t_p}\right)^{0.25}$ but not less than 0.64.

M6.10 Increased tensile strength

Unwelded metal may have increased tensile strength (R_m) depending on the surface finish.

S_r may be multiplied by the following strength factor:

A2

$$\left[1+0.5\frac{\log_{10}(N)}{6.3}\left(\frac{R_m}{400}-1\right)\left(\frac{10}{R_z}\right)^{0.25}\right] \quad \text{. . . M6.10}$$

where the maximum value of

R_m = 1000 MPa

R_z = the surface roughness (peak to valley), in µm

N = number of cycles

Typically values of R_z are:

normal plate surface = 200 µm

normal machined surface = 50 µm

ground, notch-free of notches = 10 µm

M7 DETAILED FATIGUE ANALYSIS

M7.1 Analysis required

For vessels not complying with requirements of Paragraph M4.1 a detailed fatigue analysis shall be carried out in accordance with this Paragraph M7 or an agreed equivalent.

M7.2 Notation

As in Paragraph M4.1.

M7.3 Method for detailed fatigue analysis

The method shall be as follows:

(a) For those structural discontinuities remote from weld toes that have defined geometries (smooth internal corners of known radii) the stress range, S_r, shall be determined by the use of theoretical stress concentration factors, linear elastic finite element analysis, experimental strain gauge studies or by other established means suitable for determining localized peaks in stress distributions, and the relevant fatigue curve in Figure M3 used to determine the permissible number of cycles, N.

    Strain ranges measured using strain gauges shall be converted to stress ranges using Young's Modulus prior to comparing with the fatigue strength curves in Figure M1 even if those strains indicate stresses above the yield strength.

(b) For those structural discontinuities such as weld toes, that effectively are singularities not amenable to theoretical or experimental analysis, determine the stress range distribution close to the singularity as shown in Figure M5 (between 0.4 and 3 plate thicknesses from the singularity), and by extrapolation from that distribution determine the magnitude of the stress range, S_r, at the singularity. The stress range derived from such extrapolation is called the geometric stress range. To determine the permissible number of cycles, N, the fatigue curve for 'welded' material in Figure M1 is used.

(c) If there is more than one magnitude of stress cycle then the combined effect of these cycles shall be determined by calculating the Cumulative Damage Factor U using Equation M4.4.

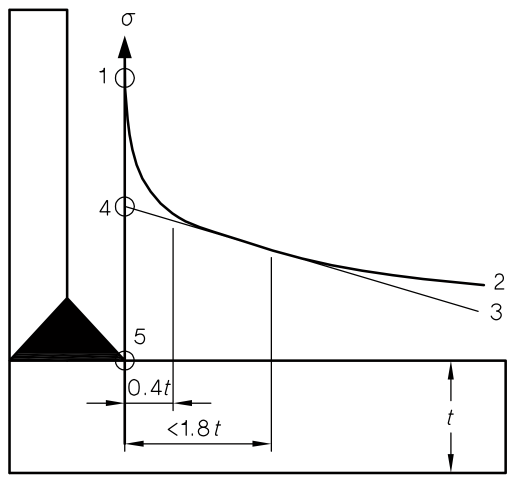
(d) For partial penetration welds, fatigue cracking through the weld starting from the root requires consideration (in addition to fatigue cracking starting from the weld toe). The procedure is to use the average stress intensity determined across the weld throat further applying a stress concentration factor of 2 to this value before determining N from Figure M1. The calculated stress range shall be determined as the calculated load divided by the weld throat area as follows:

$$\text{Calculated stress range} = \sqrt{\left(2\sigma^2 + 4\tau^2\right)} \qquad \text{. . . M7.3}$$

where

σ = average normal stress on the weld throat

τ = average shear stress on the weld throat



LEGEND
1 = Cyclic (primary + secondary + peak) stress at the weld toe.
2 = Cyclic (primary + secondary) stress distribution on the surface remote from the weld toe
3 = A straight line fitted through points at 0.4t and <1.8t on the cyclic (primary + secondary) stress distribution (2)
4 = The geometric stress is obtained by extrapolating the fitted line (3) to the weld toe
5 = The weld toe

NOTES:

1 Due to the small and highly variable root radius at the toe of the weld the peak stress component cannot be reliably determined.

2 The size of the finite element mesh or gauge length of strain gauges used to determine the stress distribution on the surface close to the weld toe should not exceed 0.2t.

FIGURE M5 GEOMETRIC STRESS

M8 FATIGUE ANALYSIS OF BOLTS

M8.1 Need for fatigue analysis

Unless the vessel on which they are installed satisfies all the conditions of Paragraph M4.1 and thus requires no fatigue analysis, the suitability of bolts for cyclic operation shall be determined in accordance with this Paragraph (M8).

M8.2 Methods of fatigue analysis

Bolts made of materials that have specified minimum tensile strengths of less than 686 MPa shall be evaluated for cyclic operation in accordance with Paragraph M4 using the applicable design fatigue curves of Figure M1 and an appropriate stress concentration factor.

High strength alloy steel bolts and studs may be evaluated for cyclic operation in accordance with Paragraph M4 using the design fatigue curve of Figure M6 provided the following requirements are complied with:

(a) The material is within the following limits:

Chromium	0.8% to 1.15%
Nickel	1.65% to 2.00%
Molybdenum	0.20% to 0.65%
Specified yield strength (R_e)	539 MPa to 981 MPa
Specified tensile strength (R_m)	686 MPa to 1128 MPa

(b) The maximum value of the service stress at the periphery of the bolt cross-section complies with Paragraph M8.3. The actual direct tension stress (averaged across the bolt cross-section) produced by the combination of preload, pressure, and differential thermal expansion, but neglecting stress concentrations, shall not exceed 2f.

(c) Threads shall be of 'V' type, having a minimum thread root radius not less than 0.075 mm.

(d) Fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

Unless it can be shown by analysis or test that a lower value is appropriate, the fatigue strength reduction factor used in the fatigue evaluation of threaded members shall be not less than 4.0.

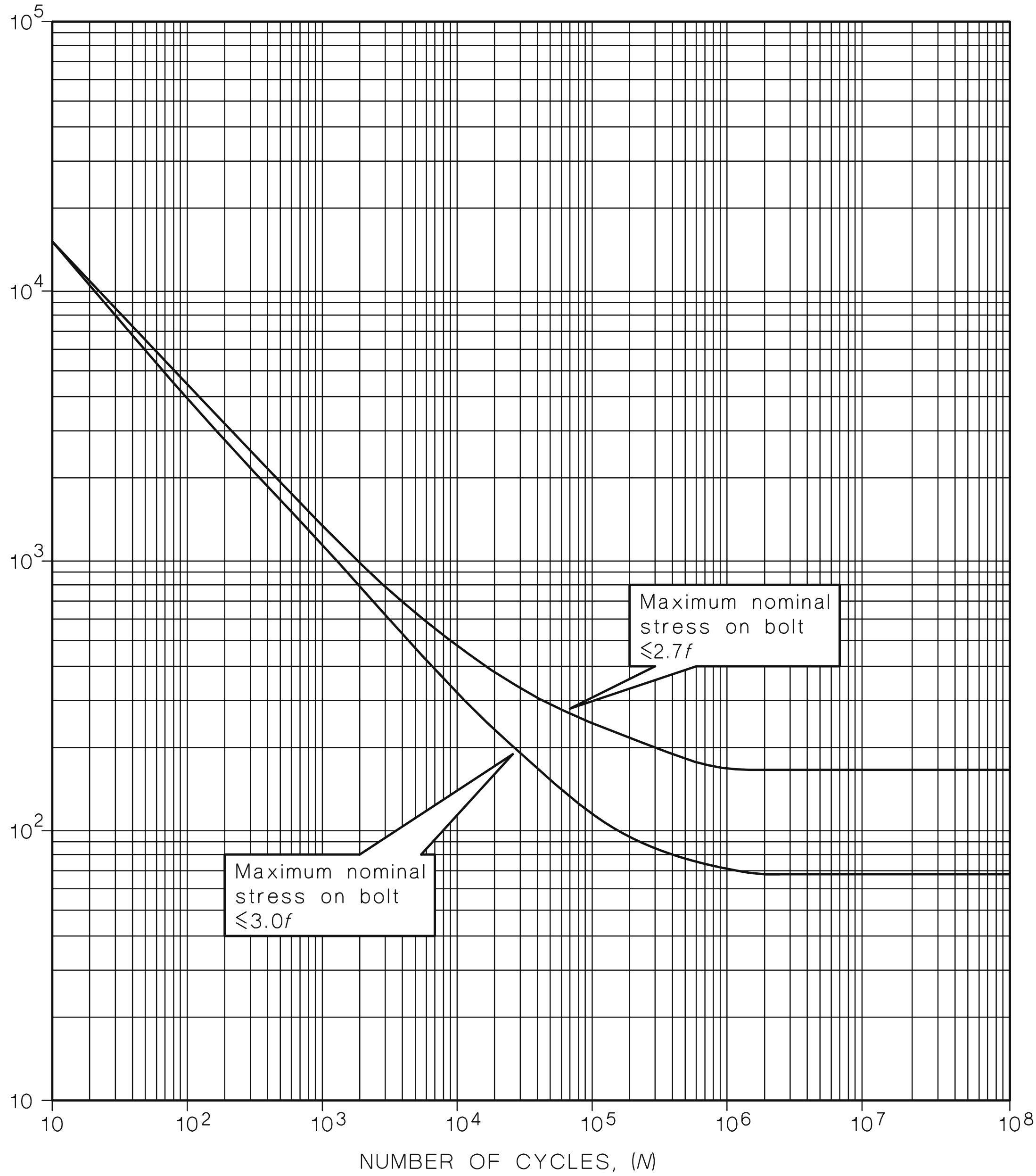


FIGURE M6 DESIGN FATIGUE CURVES FOR HIGH STRENGTH STEEL BOLTING FOR TEMPERATURES ≤375°C

M8.3 Curves for high strength bolting

The curves in Figure M6 allow for maximum effects of peak stress.

Nominal stress on the bolt is the tensile stress (primary stress) plus bending stress (secondary stress) and excludes the peak stress due to the thread root stress concentration.

This combination of tensile stress and bending stress in the bolt shall not exceed 2.7f if the upper curve is used for fatigue analysis, and shall not exceed 3f if the lower curve is used.

The tensile stress and bending stress in the bolt threaded cross section is calculated from the tensile force, bending moment and 'stressed area' of the bolt cross section:

The diameter of the 'stressed area' = $(d - 1.23p)$ . . . M8.3

where

d = the maximum diameter of the threads

p = the pitch of the threads.

M8.4 Cumulative damage

The bolts shall be acceptable for the specified cyclic application of loads and thermal stresses provided the cumulative usage factor, U, as determined in accordance with Paragraph M4.4 does not exceed 1.0.

M9 ALTERNATIVE METHODS OF CYCLIC LIFE DETERMINATION

M9.1 General

The fatigue life curves given in Paragraph M4 are based on cyclic strain data (for steels) multiplied by a Young's Modulus of 200 GPa. Accordingly those stresses quoted in Figure M1 above yield do not exist in the physical component or test specimen, but are to be understood as a quasi-elastic cyclic stress that can be used in conjunction with the results of linear elastic stress analyses.

There are unique such curves giving cyclic stress range (S_r) versus life in cycles (N), for each grade of each metal. Further for any one grade of one metal there are a range of S_r/N fatigue curves for each magnitude of mean stress.

Fatigue test data generally is subject to statistical scatter in the test results.

The fatigue curves given in Figure M1 are intended to be sufficiently conservative such that they are safe to use regardless of the metal, mean stress and statistical scatter in the source data. It should be noted however that these curves may in some instances be non-conservative, for example in corrosive environments.

It is to be understood that fatigue/life predictions are intended to be supplemented by in–service inspections, particularly toward the end of the design life, and that such inspections can be expedited by the recording and record keeping of those locations (generally few in number and highly localized) on any one vessel of 'hot spot' stresses determined at the design and stress analysis stage.

It is permissible to use a less conservative fatigue life curve if either of the following conditions apply:

(a) Established and recognized fatigue data is available in the literature for the material that is relevant with respect to—

   (i) its grade, temper and any other parameter than might affect its fatigue life;

   (ii) the stress regime including stress range, mean stress and residual stress; and

   (iii) the design stress range used is no more than 70% of the mean test data stress range.

(b) The fatigue performance of the vessel or vessel component can be established by cyclic testing.

M9.2 Fatigue life determined by testing

The test specimens, test vessel or test vessel component shall be constructed of the same material in the same condition (e.g. same mechanical working, heat treatment, surface finish etc.) as the vessel design being evaluated for fatigue life.

At least three specimens shall be tested using cyclic loading having the same ratio of mean load to range in load (as the vessel design being evaluated for fatigue life) until failure, failure being defined as fracture through a solid section that would result in fluid leakage.

The cyclic rate of testing shall not result in appreciable heating of the specimen and shall not exceed 100 Hz.

The magnitude of test load range should be set to give, as far as practical, as large a range in number of cycles to failure as possible but in no case less than 1000 cycles.

If fatigue testing is carried out at a test temperature not equal to the design temperature then the test load range shall be corrected for the difference in Young's Modulus as follows:

(a) Test load range = design load range × (E_{test}/E_{design}).

(b) Plot the test results on a graph of the form of Figure M1.

(c) Best fit through the graphed test results a curve of the form—

$$N = B/L^3 \text{ for } N < 5\,000\,000 \text{ cycles} \quad \ldots \text{M9.2(1)}$$

$$N = C/L^5 \text{ for } N > 5\,000\,000 \text{ cycles} \quad \ldots \text{M9.2(2)}$$

where

B and C are constants determined from the test results

L is the test load range × (E_{design}/E_{test}).

Values of load used in design shall not exceed 70% of those determined by Equations M9.2(1) and M9.2(2).

M10 DETERMINATION OF FATIGUE STRENGTH REDUCTION FACTORS

The following criteria shall be applied in the determination of fatigue strength reduction factors:

(a) A reduction in fatigue strength of a component can be due to the presence of a notch, (a 'notch' for the purpose of this Appendix being an actual notch or an abrupt change in cross-section), or a transition section of differing curvatures, or attachments for supports, or penetrations into shell, e.g. drill holes and welded nozzles with varying diameters and corner radii.

(b) The fatigue strength reduction factor shall be determined by tests on 'notched' and 'unnotched' specimens and calculated as the ratio of the 'unnotched' stress to the 'notched' stress for failure or other equivalent methods.

(c) The test part shall be fabricated from the same material and shall be subjected to the same heat treatment as the component.

(d) The stress level in the specimen shall not exceed the limit given in Appendix H, and shall be such that failure does not occur in the less than 1000 cycles.

(e) The configuration, surface finish, and stress state of the specimen shall closely simulate those expected in the components. In particular, the stress gradient shall not be more abrupt than expected in the component.

(f) The cyclic rate shall be such that appreciable heating of the specimen does not occur, nor shall it exceed 100 Hz.

M11 CYCLIC THERMAL STRESSES

Where a vessel is subject to cyclic thermal stress conditions, the following recommendations should be considered:

(a) Pressure vessels which operate at elevated or subzero temperature should be heated or cooled slowly, and should be efficiently lagged to minimize temperature gradients in the shells. Rapid changes of shell temperature should be avoided.

(b) The vessels should be able to expand and contract without undue restraint including restraint on differential thermal expansion.

Where the recommendations in Items (a) and (b) above are not observed and the vessel does not satisfy Equation M4.1, thermal stresses due to temperature changes shall be considered.

The use of pad-type reinforcement or partial penetration joints is not suitable for cases where there are significant temperature gradients, especially where these are of a fluctuating nature.

M12 VIBRATIONS

Pressure pulsations, turbulent or flashing flow, wind excitation or forces transmitted from plant (e.g. rotating or reciprocating machinery), may cause vibration of piping or pressure vessel shells. Where vibration occurs, the likelihood of failure may be reduced by mechanical detailing, that is by reducing overhung masses, reducing stress concentrations, etc. Where vibration fatigue is explicitly considered in the design phase, an infinite fatigue life should be used, as the number of vibration cycles may be extremely large. In most cases these vibrations cannot be quantified at the design stage. Therefore an examination of plant should be conducted following initial start-up. Where excessive vibration occurs, the source of vibration should be eliminated, or be isolated by stiffening the additional support or adding damping, introduced at a location local to the vibration.

M13 CREEP FATIGUE

For cyclic loading in the creep range, creep-fatigue interaction shall be assessed as agreed by the parties concerned. (See AS/NZS 3788 and references, ASME BPV-III, UK-R5).

M14 CASTINGS

For cast components that require detailed fatigue analysis, excluding grey cast iron, the fatigue strength shall be established by one or more of the following methods:

Method 1—

(a) Determine the peak stress locations on the casting.

(b) Use NDE to confirm any casting flaws in the peak stress locations do not exceed in size and frequency those that would be acceptable in welds as detailed in AS 4037.

(c) Determine the permissible cyclic life from the ‘welded’ curve in Figure M1.

Method 2—

(i) Determine the peak stress locations on the casting.

(ii) Determine, using fracture mechanics, the maximum permissible flaw size that would result in a fluctuation in crack tip stress intensity less than the threshold stress intensity (≤5 MPa√m for steels).

(iii) Use NDE to ensure there are no flaws in the peak stress locations greater in size than the maximum permissible flaw size determined in (b) above.

Method 3—

Determine the fatigue strength using the methods for castings given in one or more of the following Standards:

(A) AS/NZS 3788, Appendix O.

(B) API 579.

(C) BS 7910.

(D) ASME-BPV-III.

APPENDIX N

DESIGN FOR LOCAL NOZZLE AND STRUCTURAL NON-PRESSURE LOADS

(Normative)

N1 GENERAL

This Appendix sets out methods for the evaluation of external (non-pressure) loads on pressure vessels, and acceptance criteria for the resulting local stresses. The use of other methods that provide an equivalent level of safety is the subject of agreement between the parties concerned.

NOTES:

1 The requirements in this Appendix are additional to those of Clauses 3.18 (for openings) and 3.19 (for nozzles), and provides guidance in compliance with Clauses 3.24 (for supports) and 3.25 (for attached components and structures).

2 An increase in component thickness or the addition of reinforcing pads might be required to account for external loads acting concurrently with the pressure loads on the pressure envelope.

External loads are those applied by any source other than from the pressure (internal or external) load for the vessel, for example those applied by connected piping, structural attachments, or support reactions.

N2 SPECIFICATION OF EXTERNAL LOADS

The types and sources of external loads to be considered shall include (but are not limited to) the following:

(a) Loadings as defined in Clause 3.2.3 (where appropriate).

(b) Loads specified to the designer (e.g. by the purchaser), including local loads (e.g. specified nozzle loads), platform and pipe support loadings and other substantial structural loads (e.g. due to saddles or supports). Reference to Clause 3.7.5 may also be required.

(c) Other loads defined in the design documentation or as agreed between the parties concerned.

(d) The resultant of all local loads and global environmental loads at the vessel supports, for support design in accordance with Clauses 3.24 and 3.25 and with this Appendix.

NOTE: The need for any limits on piping loads which are to be applied to the vessel design, should be specified to the designer. Application of such limits might require discussion and agreement between the parties concerned.

All external loads credible under normal service, start-up, shut-down and upset conditions shall be considered. Where required by the purchaser or this Standard, the design shall allow for the effects of these loads on the vessel design with or without pressure as appropriate.

NOTE: Vessels with large D/T ratios generally have a low inherent capacity for local loading unless additional reinforcement or strengthening is provided over and above that required for pressure.

The localized loading at supports, such as saddles, skirts, discrete support lugs and ring beams, need to consider the loads described above plus the vessel weight, friction effects (when applicable, e.g. at sliding supports) and eccentricity to the shell of the supports, to ensure that localized stresses produced from these loads do not exceed the acceptance criteria specified in this Appendix.

Where external loads for nozzles and attachments are specified to the designer, these shall be reviewed to ensure they are compatible with the specified vessel design parameters. Where specified loads are not compatible with the rest of the specification, the design external loads are the subject of agreement between the parties concerned. Design external loads shall be clearly set out in the design documentation.

NOTES:

1 For example, it might not be possible to apply the nominated external loads to the vessel without the use of reinforcing pads, which might be prohibited by the specification. Also for example, nominated piping loads might exceed the capacity of the flanges between the nozzle and piping.

2 The specified direction and maximum size of local loads for localized nozzle stress calculations might not be credible when combined with all other maximum nozzle loads for the design of the total resultant piping load acting on vessel supports. Engineering studies or a detailed assessment of piping loads, directions and load cases can provide a more credible total resultant piping load. In the absence of such assessments or of any specified design loads, the supports should be designed for a minimum of 50% of the most conservative addition of all credible piping loads as applied to the vessel supports.

N3 EVALUATION METHODS

N3.1 General

Primarily there are two distinct approaches to the evaluation of localized external loads on pressure vessels. These are finite element analysis (FEA) and 'manual analytical' methods (such as PD 5500 Appendix G or WRC 107/WRC 297 and WRC 368).

Where this Appendix allows modifications to the original method, this is intended to provide more practical results where a method is known to be excessively conservative. The specified acceptance criteria are consistent with each method’s accuracy and with the basic philosophies of this Standard.

NOTE: The manual analytical methods outlined in this Appendix differ in the locations of calculated stresses. PD 5500 Appendix G calculates the maximum stress in 4 defined quadrants, whereas the WRC methods calculate the stress at 8 points on the inside and outside surfaces of the longitudinal and circumferential meridians. Consequently, a direct comparison between the different methods is not possible.

N3.2 Evaluation using past experience

Previous successful experience or conservative engineering judgement may be used to evaluate the effects of external loads, in which case the basis of the experience or judgement shall be clearly documented. Documentation shall include consideration of any differences in design, materials, application and other relevant issues (e.g. changes to safety factors on design strengths).

NOTE: The use of past experience or conservative engineering judgement should be the subject of discussion and agreement between the parties concerned.

N3.3 Evaluation using finite element analysis (FEA)

Where FEA analysis is used, it shall be undertaken in accordance with Appendix I. Results shall be evaluated in accordance with Appendices H and I.

NOTE: The requirements in Appendix N are intended for manual analytical methods, and are unsuitable for use with FEA.

N3.4 Evaluation using PD 5500, Appendix G

PD 5500, Appendix G may be used to evaluate the effects of external loads, provided it is used in its entirety, and provided that the specified limitations are complied with.

Values may be extrapolated beyond the specified curves, provided that such extrapolation is conservative.

Where a reinforcing pad or insert plate is used at a nozzle, the pressure stresses at the edge of the nozzle shall be calculated using the combined thickness of the shell and pad, or the insert plate thickness provided that full penetration welds are used for the connection. Reinforcing plates used with solid attachments (i.e. without a hole in the shell) are not considered to carry any pressure stress.

Where an attachment is only connected to the reinforcing pad (rather than both the reinforcing pad and the vessel shell) then the reinforcing pad must be of adequate thickness to transfer any load to its outer edges without considering any benefit from the shell. It recognised that for loadings that produce compressive loads on the shell the shell will carry some load. However the rule of PD 5500 generally cannot make this distinction and hence conservatism requires the benefit of the shell to be ignored unless FEA or other design approaches are utilised. Generally for this design calculation, classical plate stress analysis theory (refer to Timoshenko and Roark texts as an example, see footnotes to Appendix I for bibliographic details) will need to be used to determine the reinforcing pad thickness and PD 5500 will need to be used to determine shell stresses at the pad edge.

NOTE: It cannot be assumed that the combined thickness of the shell and pad share the load.

Stress limits shall be as set out in PD 5500.

N3.5 Evaluation using WRC 107 with WRC 297

The WRC methods shall not be used beyond their limits of applicability (such as diameter to thickness ratios). WRC 107 should not be used for nozzles in shells (i.e. attachments where the shell is pierced), but is suitable for non-pressurized attachments, and for nozzles in ends. WRC 297 may be used for pressurized nozzles in shells.

The WRC 297 method is suitable for the calculation of stresses due to external loads on nozzles.

NOTE: The WRC 297 method is generally considered to be overly conservative in the calculation of stresses in the nozzle wall.

The methods of WRC 297 may be modified by calculating the resultant maximum tensile stress in the nozzle wall due to pressure and the local combined bending moment, and comparing this stress against a maximum of $1.2 \times f$ for the nozzle at design temperature.

NOTES:

1 The WRC 107/297 methods do not consider the effects of any pressure stress intensification factor (PSIF) that may apply at an opening. The use of WRC 107/297 for nozzles specifically requires the stresses due to pressure to be calculated in the shell or head, an appropriate PSIF to be applied to this stress, and the total pressure stress added algebraically to the calculated stress from the external load at the nozzle OD.

2 Previous industry approaches of adding ‘pressure thrust’ to the radial load, adding the hoop stress to the calculated membrane stress, etc. are not considered appropriate for use with current design strengths.

When a reinforcing pad or insert plate is used, the pressure stresses at the edge of the nozzle shall be calculated using the combined thickness of the shell and pad, or the insert plate thickness. Reinforcing plates used with solid attachments (i.e., without a hole in the shell) are not considered to carry any pressure stress and thus do not contribute to a reduction in the shell stresses locally. This stress shall be multiplied by the calculated PSIF and added to the stress due to the external load.

The PSIF may be determined using PD 5500, WRC 368, ASME BPV-VIII–2 (2004) Table AD560.7, or another suitable method that is agreed between the parties concerned.

NOTES:

1 Selection of the most appropriate PSIF source is the subject of agreement between the parties concerned.

2 The various methods have different levels of accuracy and conservatism. However, PD 5500 appears to be the least conservative method for determining the PSIF.

An acceptable modification to the determination of shell stress resulting from pressure is to only apply the PSIF as calculated by PD 5500 to the circumferential membrane pressure stress in the longitudinal plane, and the PSIF to the longitudinal membrane pressure stress in the circumferential plane.

The following stress limits apply:

(a) The maximum calculated membrane stress shall not exceed $1.2 \times f$.

(b) The maximum calculated shear stress shall not exceed $0.5 \times f$.

(c) The maximum calculated local membrane + bending stress shall not exceed the following:

 (i) $1.5 \times R_{eT}$ at the edge of a nozzle.

 (ii) $1.3 \times R_{eT}$ at the edge of a solid attachment or edge of a reinforcing pad, where pads are within a radius of $0.5 \times (R_T)^{0.5}$ from the OD of the nozzle or 0.15 of the shell circumference from the nozzle centreline.

 (iii) $1.5 \times f$ at the edge of a solid attachment or edge of a reinforcing pad, where pads are outside a radius of $0.5 \times (R_T)^{0.5}$ from the OD of the nozzle or 0.15 of the shell circumference from the nozzle centreline.

(d) Compressive membrane stresses shall not exceed $0.9 \times R_{eT}$.

N3.6 Evaluation using other manual analytical methods

Other manual analytical methods, including those set out in overseas pressure equipment standards or authoritative reference texts, may be used where they are based on sound technical principles and where the method of evaluation of the results provides an equivalent level of safety to this Standard.

NOTE: The use of other manual analytical methods should be the subject of agreement between the parties concerned.

Where such methods are used, the following shall be considered:

(a) Any assumptions or limitations of the method.

(b) Any restrictions on the applicability of the method.

(c) The use of appropriate safety factors to ensure an appropriate level of conservatism.

NOTE: This is particularly applicable where a traditional method is to be combined with modern design strengths based on reduced safety factors.

Acceptance criteria for stresses shall not exceed those set out in this Appendix.

APPENDIX O
NOT ALLOCATED

APPENDIX P

CONSTRUCTION BODIES AND PERSONNEL

(Informative)

P1 GENERAL

This Standard refers to various bodies and personnel whose competency to perform various functions or tasks is significant in ensuring that pressure vessels comply with this Standard and can perform satisfactorily and safely when properly used as intended.

This Appendix lists these bodies and personnel and their intended functions, and provides guidance on the assessment of competence in each role.

P2 BODIES, PERSONNEL AND FUNCTIONS

This Standard (AS 1210) generally adopts the practice shown in Table P1.

TABLE P1

BODIES AND PERSONNEL

Bodies	Personnel	Functions
Purchaser (includes owner, user or agent)		Specify vessel requirements including conformity assessment, delivery and contract
Manufacturer (assumes designer and fabricator are included)		Construct and supply vessel to contract, with MDR
Designer (body or an individual)		Design vessel to specification and standard
Material supplier		Supply required materials and components
Fabricator		Manufacture vessel to design and contract
Fabrication personnel include:	Production manager Welding engineer/supervisor Welders Examiners and testers	Control of manufacture Control and supervision of welding Perform welding Examine and perform destructive and non-destructive testing
Test body		Perform destructive and non-destructive testing
Inspection body – manufacturer	Quality manager and inspectors	Inspect, control and ensure work quality
– purchaser	Design verifier or checker Fabrication inspector	Verify design to specification and standard Inspect work to specification and standard
– independent third party	Design verifier or fabrication inspector	As for purchaser inspection body
Quality system certifying body		Certify quality systems
Accreditation body		Accredit inspection and test body

P3 Guidance of competency

Table P2 provides guidance for the assessment of competency, listing some examples of possible evidence to be considered. The assessment of competency should consider a combination of different training, qualifications and experience which is suitable for the task to be performed.

TABLE P2

COMPETENCY GUIDE

Bodies	Possible evidence of competency
1 Manufacturer (including fabricator)	1.1 Has satisfactory knowledge and experience in the particular or similar type of vessel, design and manufacturing standards, materials, size and thickness. 1.2 Has the personnel and equipment necessary to manufacture the vessel. 1.3 Can demonstrate successful previous delivery of vessels, considering quality, time and cost. 1.4 Has a suitable management system addressing quality of manufacture, with documentation of essential elements. 1.5 Has a quality management system certified or accredited by a nationally recognized approvals organization (e.g. to AS/NZS ISO 9001, AS/NZS ISO 3834, EU Pressure Equipment Directive, ASME, or equivalent). 1.6 Where required by the regulatory authority, is acceptable in the jurisdiction where the vessel will be used.
2 Designer	2.1 Has satisfactory knowledge and experience in the design of the type of vessel. 2.2 Has a design quality management system certified or accredited by a nationally recognized approvals organization, (e.g. to AS/NZS ISO 9001, AS/NZS 3834, EU Pressure Equipment Directive, ASME, or equivalent). 2.3 Has suitably qualified personnel.
3 Fabrication personnel	3.1 Have suitable experience and training in the type of vessel construction, fabrication standards, materials and processes. 3.2 Welders are qualified in accordance with the requirements of this Standard. 3.3 Welders have an appropriate certificate to AS 1796, or qualification from the International Institute of Welding (IIW), Welding Technology Institute of Australia (WTIA), NZ CBIP or NZ Institute of Welding, or equivalent qualification.
4 Testing personnel	4.1 Have satisfactory knowledge and experience in the particular test methods. 4.2 Hold appropriate certification from an organisation such as AINDT or equivalent (e.g. for radiography and ultrasonic testing, certification to AS 3998 or ISO 9712). 4.3 The testing body is accredited to perform the relevant tests by a nationally recognised accreditation organisation e.g. NATA, IANZ or equivalent organization.
5 Inspection body	5.1 Has satisfactory knowledge and experience in the type of vessel and inspection (e.g. significant experience with commercial inspection bodies, or previous experience with a regulatory pressure equipment inspection body). 5.2 For in-service inspection, has appropriate certification from an organisation such as AICIP (in Australia), CBIP (in New Zealand), National Board BPVI (in USA), API or equivalent. Certification is mandatory under New Zealand regulations. 5.3 While AS/NZS ISO/IEC 17020 does not apply directly to personnel, it does include requirements for the expertise, training and qualifications of inspectors employed by an inspection body, and for their training and development system. These requirements are considered as part of the accreditation of inspection bodies by national bodies (see 1.2 above), and thus may provide some evidence for competence of inspectors and approved signatories. 5.4 For design verification, acceptable for registration under the systems operated by IEAust, IPENZ, or equivalent. 5.5 For fabrication and welding inspection, appropriate certification from IIW, WTIA, NZ CBIP, or an equivalent organization.

APPENDIX Q

NOT ALLOCATED

APPENDIX R

LIST OF REFERENCED DOCUMENTS

(Normative)

AS

1056	Storage water heaters
1056.1	Part 1: General requirements
1074	Steel tubes and tubulars for ordinary service
1110	ISO metric hexagon bolts and screws—Product grades A and B
1110.1	Part 1: Bolts
1110.2	Part 2: Screws
1111	ISO metric hexagon bolts and screws—Product grade C
1111.1	Part 1: Bolts
1111.2	Part 2: Screws
1112	ISO metric hexagon nuts
1112.1	Part 1: Style 1—Product grades A and B
1112.2	Part 2: Style 2—Product grades A and B
1112.3	Part 3: Product grade C
1112.4	Part 4: Chamfered thin nuts—Product grades A and B
1170	Structural design actions
1170.4	Part 4: Earthquake actions in Australia
1228	Pressure equipment—Boilers
1271	Safety valves, other valves, liquid level gauges, and other fittings for boilers and unfired pressure vessels
1275	Metric screw threads for fasteners
1349	Bourdon tube pressure and vacuum gauges
1358	Bursting discs and bursting disc devices—Application, selection and installation
1391	Metallic materials—Tensile testing at ambient temperature
1544	Methods for impact tests on metals
1544.2	Part 2: Charpy V-Notch
1548	Fine grained, weldable steel plates for pressure equipment
1565	Copper and copper alloys—Ingots and castings
1721	General purpose metric screw threads
1722	Pipe threads of Whitworth form
1722.1	Part 1: Sealing pipe threads
1722.2	Part 2: Fastening pipe threads
1796	Certification of welders and welding supervisors
1830	Grey cast iron
1831	Ductile cast iron
1832	Malleable cast iron
1833	Austenitic cast iron
2129	Flanges for pipes, valves and fittings

AS	
2205	Methods for destructive testing of welds in metal
2205.7.2	Method 7.2: Dropweight fracture toughness test for nil-ductility transition temperature
2291	Metallic materials—Tensile testing at elevated temperatures
2327	Composite structures
2528	Bolts, studbolts and nuts for flanges and other high and low temperature applications
2593	Boilers—Safety management and supervision systems
2613	Safety devices for gas cylinders
2809	Road tank vehicles for dangerous goods
2809.1	Part 1: General requirements for all road tank vehicles
2809.3	Part 3: Road tank vehicles for compressed liquefied gases
2809.4	Part 4: Tankers for toxic and corrosive cargoes
2809.6	Part 6: Tankers for cryogenic liquids
2812	Welding, brazing and cutting of metals—Glossary of terms
2865	Confined spaces
2872	Atmospheric heating of vessels containing fluids—Estimation of maximum temperature
2971	Serially produced pressure vessels
3500	Plumbing and drainage Code
3500.4	Part 4: Heated water services
3597	Structural and pressure vessel steel—Quenched and tempered plate
3600	Concrete structures
3711	Freight containers
3711.6	Part 6: Tank containers
3857	Heat exchangers—Tubeplates—Methods of design
3873	Pressure equipment—Operation and maintenance
3892	Pressure equipment—Installation
3920	Assurance of product quality
3920.1	Part 1: Pressure equipment manufacture
3990	Mechanical equipment—Steelwork
4024	Safety of machinery (series)
4037	Pressure equipment—Examination and testing
4041	Pressure piping
4100	Steel structures
4291	Mechanical properties of fasteners made of carbon steel and alloy steel
4291.1	Part 1: Bolts, screws and studs
4343	Pressure equipment—Hazard levels
4458	Pressure equipment—Manufacture
4552	Gas fired heaters for hot water supply and/or central heating

AS
4774 Work in compressed air and hyperbaric facilities
4774.2 Part 2: Hyperbaric oxygen facilities

4942 Pressure equipment—Glossary of terms

61508 Functional safety of electrical/electronic/programmable electronic safety-related systems (series)
B148 Unified black hexagon bolts, screws and nuts (UNC and UNF threads) and plain washers—Heavy series (Obsolescent)

AS IEC
61511 Functional safety—Safety instrumented systems for the process industry sector (series)

AS ISO
7 Pipe threads where pressure-tight joints are made on the threads
7.1 Part 1: Dimensions, tolerances and designation

AS/NZS
1170 Structural design actions
1170.0 Part 0: General principles
1170.2 Part 2: Wind actions

1200 Pressure equipment

1425 LP Gas fuel systems for vehicle engines

1594 Hot-rolled steel flat products

1596 The storage and handling of LP Gas

2022 Anhydrous ammonia—Storage and handling

2451 Bolts, screws and nuts with British Standard Whitworth threads

2465 Unified hexagon bolts, screws and nuts (UNC and UNF threads)

3509 LP Gas fuel vessels for automotive use

3678 Structural steel—Hot-rolled plates, floorplates and slabs

3679 Structural steel
3679.1 Part 1: Hot-rolled bars and sections

3788 Pressure equipment—In-service inspection

3992 Pressure equipment—Welding and brazing qualification

4331 Metallic flanges
4331.1 Part 1: Steel flanges
4331.2 Part 2: Cast iron flanges
4331.3 Part 3: Copper alloy and composite flanges

4692 Electric water heaters (series)

AS/NZS ISO
3834 Quality requirements for fusion welding of metallic materials (series)

9001 Quality management systems—Requirements

31000 Risk management—Principles and guidelines

AS/NZS, ISO/IEC
17020 General criteria for the operation of various types of bodies performing inspection

ISO

185	Grey cast iron—Classification
643	Steels—Micrographic determination of the apparent grain size
1083	Spheroidal graphite cast irons—Classification
2892	Austenitic cast irons—Classification
5922	Malleable cast iron
7005	Metallic flanges (Series)
9328	Steel flat products for pressure purposes—Technical delivery conditions
9328-7	Part 7: Stainless steels
16528	Boiler and pressure vessels
16528-1	Part 1: Performance requirements
16812	Petroleum, petrochemical and natural gas industries—Shell-and-tube heat exchangers
20421	Cryogenic vessels—Large transportable vacuum-insulated vessels (series)
21028	Cryogenic vessels—Toughness requirements for materials at cryogenic temperature (series)
21029	Cryogenic vessels—Transportable vacuum insulated vessels of not more than 1000 litres volume (series)

API

5B	Threading, gauging and thread inspection of casing, tubing and line pipe threads
5L	Line pipe
579	Fitness-for-service
620	Design and construction of large, welded, low-pressure storage tanks

ANSI/API

RP 520	Sizing, selection and installation of pressure-relieving devices in refineries
RP 530	Calculation of heater tube thickness in petroleum refineries

ANSI/AWS

A5.8	Specification for filler metals for brazing and braze welding

ASME

B1.20.1	Pipe threads, general purpose, inch
B1.20.3	Dryseal pipe threads, inch
B16.5	Pipe flanges and flanged fittings
B16.9	Factory-made wrought steel buttwelding fittings
B16.11	Forged fittings, socket welded and threaded
B16.34	Valves—Flanged, threaded and welding end
B16.47	Large diameter steel flanges
B31.3	Process piping
B36.10	Welded and seamless wrought steel pipe
BPV	Boiler and Pressure Vessel Code
BPV-II-D	Section II—Materials; Part D Properties
BPV-III	Section III—Rules for Construction of Nuclear Facility Components
BPV-VIII-1	Section VIII—Rules for construction of pressure vessels: Division 1
BPV-VIII-2	Section VIII—Rules for construction of pressure vessels: Division 2—Alternative rules

ASME

BPV-X Section X: Fibre-reinforced plastic pressure vessels

PVHO-1 Safety standard for pressure vessels for human occupancy

ASTM

A 53 Specification for pipe, steel, black and hot-dipped, zinc-coated, welded and seamless

A 105 Specification for carbon steel forgings for piping applications

A 106 Specification for seamless carbon steel pipe for high-temperature service

A 193 Specification for alloy-steel and stainless steel bolting for high-temperature or high pressure service and other special purpose applications

A 194 Specification for carbon and alloy steel nuts for bolts for high-pressure and high-temperature service, or both

A 203 Specification for pressure vessel plates, alloy steel, nickel

A 204 Specification for pressure vessel plates, alloy steel, molybdenum

A 216 Specification for steel castings, carbon, suitable for fusion welding, for high-temperature service

A 240 Specification for heat-resisting chromium and chromium nickel stainless steel plate, sheet and strip for pressure vessels and for general applications

A 263 Specification for stainless chromium steel-clad plate

A 264 Specification for stainless chromium-nickel steel-clad plate

A 265 Specification for nickel and nickel-base alloy-clad steel plate

A 266 Specification for carbon steel forgings for pressure vessel components

A 320 Specification for alloy steel and stainless steel bolting materials for low-temperature service

A 325 Specification for structural bolts, steel, heat treated, 120/105 ksi minimum tensile strength

A 350 Specification for carbon and low-alloy steel forgings, requiring notch toughness testing for piping components

A 352 Specification for steel castings, ferritic and martensitic, for pressure-containing parts, suitable for low-temperature service

A 353 Specification for pressure vessel plates, alloy steel, double-normalized and tempered 9% nickel

A 354 Specification for quenched and tempered alloy steel bolts, studs, and other externally threaded fasteners

A 370 Test methods and definitions for mechanical testing of steel products

A 387 Specification for pressure vessel plates, alloy steel, chromium-molybdenum

A 420 Specification for piping fittings of wrought carbon steel and alloy steel for low-temperature service

A 449 Specification for hex cap screws, bolts and studs, steel, heat treated, 120/105/90 ksi minimum tensile strength, general use

A 453 Specification for high-temperature bolting materials, with expansion coefficients comparable to austenitic stainless steels

ASTM

A 517 Specification for pressure vessel plates, alloy steel, high-strength, quenched and tempered

A 524 Specification for seamless carbon steel pipe for atmospheric and lower temperatures

A 540 Specification for alloy-steel bolting materials for special applications

A 574 Specification for alloy steel socket-head cap screws

A 789 Specification for seamless and welded ferritic/austenitic stainless steel tubing for general purposes

B 96 Specification for copper-silicon alloy plate, sheet, strip, and rolled bar for general purposes and pressure vessels

B 127 Specification for nickel-copper alloy (UNS N04400) plate, sheet and strip

B 152 Specification for copper sheet, strip, plate, and rolled bar

B 171 Specification for copper-alloy plate and sheet for pressure vessels, condensers, and heat exchangers

B 209 Specification for aluminium and aluminium-alloy sheet and plate

B 265 Specification for titanium and titanium alloy strip, sheet, and plate

B 424 Specification for nickel-iron-chromium-molybdenum-copper alloy (UNS N08825 and US N08821) plate, sheet and strip

B 574 Specification for low-carbon nickel-chromium-molybdenum, low-carbon nickel-molybdenum-chromium-tantalum, low-carbon nickel-chromium-molybdenum-copper, and low-carbon nickel-chromium-molybdenum-tungsten alloy rod

B 575 Specification for low-carbon nickel-chromium-molybdenum, low-carbon nickel-chromium-molybdenum-copper, low-carbon nickel-chromium-molybdenum-tantalum, and low-carbon nickel-chromium-molybdenum-tungsten alloy plate, sheet, and strip

B 898 Specification for reactive and refractory metal clad plate

BS

853 Specification for vessels for use in heating systems (series)

2693 Screwed studs

2693.1 Part 1: General purpose studs

3293 Specification for carbon steel pipe flanges (over 24 inches nominal size) for the petroleum industry

3799 Specification for steel pipe fittings, screwed and socket-welding for the petroleum industry

4076 Specification for steel chimneys

4439 Specification for screwed studs for general purposes. Metric series

4882 Specification for bolting for flanges and pressure containing purposes

4994 Specification for design and construction of vessels and tanks in reinforced plastics

5154 Specification for copper alloy globe, globe stop and check, check and gate valves

6374 Lining of equipment with polymeric materials for the process industries

6374.1 Part 1: Specification for lining with sheet thermoplastics

6374.2 Part 2: Specification for lining with non-sheet applied thermoplastics

BS
6374.3 Part 3: Specification for lining with stoved thermosetting resins
6374.4 Part 4: Specification for lining with cold curing thermosetting resins
6374.5 Part 5: Specification for lining with rubbers

7910 Guide to methods for assessing the acceptability of flaws in metallic structures

BSI PD
5500 Specification for unfired fusion welded pressure vessels

6510 A review of the present state of the art of assessing remanent life of pressure vessels and pressurized systems designed for high temperature service (obsolescent)

EN
1092 Flanges and their joints. Circular flanges for pipes, valves, fittings and accessories, PN designated (series)

13445 Unfired Pressure Vessels (series)

15761 Steel gate, globe and check valves for sizes D100 and smaller, for the petroleum and natural gas industries

European Parliament and European Council
PED 97/23/EC Pressure Equipment Directive (PED)

JIS
G3115 Steel plates for pressure vessels for intermediate temperature service

TEMA Standards of Tubular Exchanger Manufacturers Association, Inc.

EJMA Standards of the Expansion Joint Manufacturer's Association

Bednar Henry H *Pressure Vessel Design Handbook*, 2nd edition, Van Nostrand Reinhold Publication

Freese C.E. *Vibration of Vertical Pressure Vessels*. ASME Paper 58-PET-13 July 1958

De Ghetto, K. & Long, W., *Design Method To Check Towers For Dynamic Stability.* Hydrocarbon Processing, Feb. 1966.

Mahajan Kanti, K., *Tall Stack Design Simplified*. Hydrocarbon Processing, Sept. 1975.

Moody, G. B., *Mechanical Design To Tall Stacks*. Hydrocarbon Processing Sept 1969.

WRC
107 Welding Research Council Bulletin 107, *Local stresses in spherical and cylindrical shells due to external loading*

297 Welding Research Council Bulletin 297, *Local stresses, in spherical and cylindrical shells due to external loadings*. Supplement to WRC-107.

368 Welding Research Council Bulletin 368, *Stresses in intersecting cylinders subjected to pressure*

NOHSC Worksafe guide, *Plant design—Making it safe*

IMDG International maritime dangerous goods code

Commonwealth Department of Infrastructure, Transport, Regional Services and Local Government—Federal Office of Road Safety

ADR Australian Design Rules

APPENDIX ZZ

CONFORMANCE TO DESIGN AND CONSTRUCTION CRITERIA

(Informative)

ZZ1 GENERAL

ISO 16528-1 sets out generic performance requirements for boilers and pressure vessels. Similarly, Appendix J of AS/NZS 1200—2000 lists safety recommendations to be addressed in the design and construction of pressure equipment.

This Standard aligns with listed criteria of those Standards, as described below.

ZZ2 FAILURE MODE SUMMARY

This Standard addresses all of the 'common failure modes' listed in ISO 16528-1:2007 Clause 6.2, as well as the 'failure modes to be addressed' listed in ISO 16528-1:2007, Clause 6.3.

This Standard satisfies the design criteria listed in AS/NZS 1200:2000, Paragraph J3.

ZZ3 DETAILED TECHNICAL REQUIREMENTS CHECKLIST

Table ZZ1 compares the contents of this Standard with the requirements of ISO 16528-1:2007, and the safety recommendations listed in AS/NZS 1200:2000, Appendix J.

TABLE ZZ1

COMPARISON OF TECHNICAL REQUIREMENTS

ISO 16528-1:2007 Clause		AS/NZS 1200:2000 Appendix J Clause		AS 1210 Clause	
Materials					
7.2.1	Materials—general	J5	Materials	2.1	Material specifications
7.2.2	Specification of materials	J5.3	Documentation	2.1	Material specifications
7.2.3	Material Certification	J5.4	Compliance	2.4	Material identification
Design					
7.3.1	Design—loadings and other design considerations	J3.2.1	Design, General	3.2	Design conditions
7.3.2	Design methods	J3.2	Design for adequate strength and deflection	3.1.3	Design methods
7.3.3	Design margins	J3.1 J3.2.3	Design, General Calculation method	3.1.4	Design against failure
7.3.4	Design factors	J3.1	Design, General	3.3.1.1	Design tensile strength—General
7.3.5	Means for examination	J3.4	Means of examination	3.20	Inspection openings
7.3.6	Draining and venting	J3.5	Means of draining and venting	8.16 8.17	Drainage Vents

(*continued*)

TABLE ZZ1 (*continued*)

ISO 16528-1:2007 Clause		AS/NZS 1200:2000 Appendix J Clause		AS 1210 Clause	
7.3.7	Corrosion and erosion	J3.6 J3.7	Corrosion and other chemical attack Wear	2.5.3.3 3.1.4	Corrosion resistance Design against failure
7.3.8.1	Overpressure protection —general	J3.9	Protection against exceeding allowable limits	8.2.1	Pressure relief—general requirements
7.3.8.2	Types of devices	J3.10	Safety Accessories	8.3	Types of pressure-relief devices
7.3.8.3	Safety accessories	J3.10 J6.1	Safety accessories Requirements - Fired pressure equipment	8.1	Protective devices and other fittings —General
Manufacture					
7.4.1	Manufacture—methods	J4.1	Manufacture	4.1	Manufacture—General (Refers to AS 4458)
7.4.2	Identification of materials	J4.1.6	Traceability	4.1	Manufacture—General (Refers to AS 4458 Section 4)
7.4.3	Preparation of parts	J4.1.2	Preparation of component parts	4.1	Manufacture—General (Refers to AS 4458 Sections 5,6)
7.4.4	Welding	J4.1.3	Joining	4.2	Welded construction (Refers to AS 4458)
7.4.5	Welding procedure qualifications	J4.1.3	Joining	4.2 5.2	Welded construction (Refers to AS/NZS 3992) Qualification and test plates
7.4.6	Welder qualifications	J4.1.3	Joining	4.2.2.2; 5.2	Competence of welders (Refers to AS/NZS 3992 Section 9); Qualification and test plates
7.4.7	Welder identification			4.2	Welded construction (Refers to AS 4458 Clause 9.4)
7.4.8	Heat treatment	J4.1.5	Heat treatment	4.1; Table 1.6	Refers to AS 4458 Section 14
7.4.9	Tolerances			4.2	Welded construction (Refers to AS 4458)
Inspection and Examination					
7.5.1	Inspection and examination (I&E)—general	J4.2	Final assessment	4.1 5.3	Manufacture—General Non-destructive examination (Refers to AS 4458, AS 4037)
7.5.2	I&E methods	J4.1.4	Non-destructive tests	5.3	Non-destructive examination (Refers to AS 4037 Sections 3,7,11,13)
7.5.3	I&E procedures			5.3	Non-destructive examination (Refers to AS 4037 Section 3)`
7.5.4	I&E personnel qualification	J4.1.4	Non destructive tests	5.3	Non-destructive examination (Refers to AS 4037 Section 4)
7.5.5	Evaluation of indications and acceptance criteria			5.3	Non-destructive examination (Refers to AS 4037 Sections 8,9)
7.5.6	Disposition of unacceptable imperfections	J4.1.7	Repairs	5.3	Non-destructive examination (Refers to AS 4037 Section 9)

(*continued*)

TABLE ZZ1 (*continued*)

ISO 16528-1:2007 Clause		AS/NZS 1200:2000 Appendix J Clause		AS 1210 Clause	
7.6.1	Final inspection	J4.2.2	Final inspection	4.1	Manufacture—General (Refers to AS 4458 Sections 20,21)
76.2	Final pressure test	J4.2.3	Proof Test	5.10	Hydrostatic tests (See also AS 4307)
7.7	Marking/labelling	J4.3	Marking and labelling	7.1	Marking
Conformity Assessment					
8	Conformity Assessment	J4.2.2	Final inspection	6	Conformity assessment (Refers to AS 3920.1)

INDEX

AMENDMENT CONTROL SHEET

AS 1210—2010

Amendment No. 1 (2013)

REVISED TEXT

SUMMARY: This Amendment applies to Clauses 2.6.3.2, 3.19.1, 3.19.2, 3.21.5.4.2, 3.21.6.2, 3.26.3.4, Figures 3.15.1 and 3.21.6.2 and Appendices B, H, I, J, L and M.

Published on 25 November 2013.

Amendment No. 2 (2015)

REVISED TEXT

SUMMARY: This Amendment applies to the Clauses 1.8.29 (new), 2.6.4, 3.2.5, 3.18.7.3, 3.19.10.2, 3.21.1, 3.21.6.4.1, 3.21.9.1, 3.26.3.6, and 3.26.3.8 (new), Tables 2.6.4, 3.26.3.8.1 (new) and 3.26.3.8.2 (new), and Appendices B, H, I, L and M.

Published on 28 July 2015.

NOTES

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NOTES

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ISBN 978 0 7337 9687 6

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